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XXVIII.

FAN BLOWERS.

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The present almost universal use of rotary fans for forcing combustion on naval vessels and in the merchant marine has made information in regard to the fan blower of value to the marine and naval engineer; but this information is not easily attainable, as the literature on the subject is not large, and what exists is in a fragmentary form, to be culled from the proceedings of different societies. The lack of published data in regard to the fan blower is soon made apparent to one who has occasion to investigate the subject, and inquiry as to the proper proportioning of this machine will reveal a surprising difference of opinion and of practice.

There is probably no machine in which more power is unwittingly wasted than in the fan blower, and in regard to which more misapprehension exists. The writing of specifications is common in which it is impossible to fulfil the conditions from the fact that, the revolutions of the blast wheel and its dimensions being fixed, the desired volume and pressure of blast cannot be secured. The laws governing the operation of fan blowers

are few and simple so far as practical work is concerned, and results can be predicted for them with as much certainty as for other machines.

The blast wheel of a rotary fan consists of a number of tubular passages rotating about an axis at right angles to the direction of the passages, causing air impelled by centrifugal force to issue from the ends of the passages with a velocity equal to the velocity of rotation of the extremity of the passages.

The reason for this was first pointed out by M. Peclet, who said:

"It is demonstrable in mechanics that if a tube, *A B*, cylindrical and completely open at its two ends, turns round a line, *O*, perpendicular to its axis, the air escapes by the extremity *A* with a velocity equal to the velocity of rotation at this extremity. The velocity of escape is the same whether the tube is more or less long, provided the distance *A O* remains constant; it is still the same when the tube is curved. If we suppose that the orifice *A* be closed, in all cases possible, the pressure exerted by the air against the interior surface of the partition will be equal to the height of air corresponding to the velocity of rotation of the point *A*."

From this it appears that in the case of a tube closed at the axial end and open at the periphery, the partial vacuum will be that due to the velocity of a body of the density of the atmosphere in which the tube is revolving, and also in case of the tube closed at both ends, for any density of the enclosed atmosphere, the pressure on the axial end would be less than on the opposite end by an amount due to the velocity of a body of the density of the enclosed air falling with the velocity at which the outer end of the tube moves.

What has been stated above shows that whatever the shape of the fan blades, the maximum pressure or vacuum produced by them calls for no delivery, and with a rotary fan proportioned with the case so that no regurgitations of the air occur, no power will be consumed by the fan when the inlet and outlet, or both, are closed, for no work is done. The idea is frequently met that the maximum power is consumed when the outlet is closed, and the

inquiry is often put as to the advisability of using a safety valve in the blast pipe to prevent rupture when all outlets are closed. It is sometimes difficult to convince people using fan blowers that comparatively no power is used when the outlet is closed, and that the freer the outlet for the escape or the inlet for the entrance of air, the greater will be the power consumed, they holding the reverse to be the case.

As the pressure exerted by air issuing from a blast wheel is equal to the height of a column of air represented by the velocity of rotation of the tips of the fan blades, the pressure for any given velocity, and *vice versa*, and the horse-power in the blast wheel, where

p = pressure in lbs. per sq. ft.
v = velocity of air in ft. per second.
g = gravity.
w = weight in pounds of a cu. ft. of air.
A = area through which the air moves with pressure p and velocity a.
$H.P.$ = horse-power in the blast, may be calculated from

$$v^2 = 2gh. \tag{1}$$

$$h = \frac{p}{w}. \tag{2}$$

Substituting value of h in (1), we have

$$v^2 = \frac{2gp}{w}. \tag{3}$$

$$\text{Hence } p = \frac{wv^2}{2g}. \tag{4}$$

$$\text{Now } H.P. = \frac{60\ pav}{144 \times 33{,}000} = \frac{pav}{79{,}200}. \tag{5}$$

Substituting value of p from (4) in (5), we have

$$H.P. = \frac{wav^3}{158{,}400g}. \tag{6}$$

Hence it appears from (5) and (6)

$$p \propto wv^2. \tag{7}$$

$$H.P. \propto awv^3. \tag{8}$$

That is: first, the pressure of air varies directly as the second power of the velocity and as the weight of the air, and consequently inversely as the absolute temperature, and directly as the height of the barometric column and hygrometric condition of the air; second, the power in the blast varies directly as the third power of the velocity and as the weight of the air, and consequently inversely as the absolute temperature, and directly as the weight of the barometric column and hygrometric condition of the air.

It is evident from the above that although the speed at which the air moves for any peripheral speed is constant, and also the volume of air delivered per revolution is constant, yet the weight of air moved per revolution varies with the thermometer, barometer and hygrometer. As the power in the blast, the speed remaining the same, varies in the same manner, and as the readings on these instruments vary greatly in very short times, it is evident that with any rotary fan, the speed remaining the same, the pressure produced and power consumed will vary greatly in the course of a few hours. With rotary fans driven by direct connected engines, an elevation of temperature will immediately be shown by an increase of speed, due to decreased resistance. This increase of speed is particularly noticeable with fans driven by direct connected engines located above boilers, and moving air of high temperature.

As the amount of power consumed varies directly as the third power of the velocity of the air, or peripheral speed of tips of fan blades, this peripheral or tip speed of blades should be kept as low as possible and perform the work properly. It is well to select a fan for any given work with sufficient capacity, so that the tip speed of blades will produce the desired pressure at points of delivery, plus loss due to frictional resistance between the fan and such points. A fan of inadequate capacity may be made to do the work by speeding it up, but only at a comparatively enormous expenditure of power. This speeding up of small fans of inadequate capacity for the work to which they are applied is one of the most common errors in the application of rotary fans.

It has been observed in experiments with fan blowers that the velocity of air issuing from the fan case most nearly approximated the tip speed of the blades when the area of opening through which the air was discharged was equal to the area of the blade; but in extended experiments, using wheels of larger width at tips of blades, the diameter of blast wheel remaining the same and the same casing being used, nearer equalization of the velocities of the tips of the blades and those due to the observed pressure of the air were obtained when the discharge opening was equal to the diameter of the wheel multiplied by one-third the width of the wheel at circumference.

The capacity of a rotary fan to produce pressure, or force of suction, the one being the reverse of the other, may therefore be expressed by a function of the product of the diameter and width of tips of blades of the rotating fan or blast wheel, and every wheel has a limit of capacity for producing pressure or suction over a given area, this area in square inches being $\frac{DW}{3}$ where D and W represent respectively the diameter and width at tips of blades of blast wheel, both dimensions being in inches. If the peripheral speed of a wheel is that required to produce a pressure of two ounces to the square inch, this pressure will be found on an area represented by $\frac{DW}{3}$. If this area be increased the pressure will fall off, and if it be decreased no elevation of pressure follows, but remains constant even when the outlet is closed.

In taking up any problem where it is required to move any given volume of air at a given velocity, a value is first found for the expression $\frac{DW}{3}$ by dividing the required volume in feet per minute by the cubic feet of air which will escape per minute through one square inch of opening with allowance for the *vena contracta*, this quantity being taken from a table or calculated, interpolations being readily made from a table by remembering that the volume varies directly as the velocity and the pressure as the second power of the velocity. Having found a value for $\frac{DW}{3}$

the next point to determine is whether a wheel of large diameter and narrow at tips of blades is to be used, driven at a low rotative speed, or whether a small wheel, wide at tips of blades is to be employed, driven at high speed.

As the power consumed in moving air varies as the third power of the velocity at which the air is moved, it is apparent that in designing any fan system economy is served in moving the air at the lowest admissible velocity and in using a fan of ample capacity, as nothing is lost by using a large fan, while the loss is great where too small a fan is used, necessitating, in order to obtain the requisite volume, a tip speed of blades producing a pressure in excess of the pressure of blast required to do the work.

Many forms of blades are met with in practice, but the straight radial blade is most commonly used, for the reason that the straight blade is as efficient as one that is curved, so far as the delivery of any given volume of air at a stated pressure is concerned, as the velocity with which the air is moved is solely dependent upon centrifugal force, and this force is independent of the shape of the blade. Theoretically a curve should be given to the blade which will enable the air to be received and delivered without shock, and the curvature of the blade is determined to a great extent by the velocity at which the blade will move, but no practical difference in efficiency is found between wheels with radial blades and those with curved blades, and cases are frequently met with where parties having wheels with curved blades are running them with concave face of blade advancing, claiming that the best results are secured by so doing.

A blade with a backward curve at tip will run with less noise than will a straight blade, but as stated above, so far as volume of air delivered and pressure of blast produced are concerned, the straight blade is equally as efficient as the curved blade, and so far as power to operate them is concerned it will require a careful test to detect the difference.

It is the American and English practice to turn the blast wheel when curved blades are used so that the convex surface of the blade advances toward the outlet, but the French advance

the concave surface, and the blast wheels of their exhausting fans have blades with sharp backward turn at tips. There is also a marked difference between the French and ourselves as to the relative proportions of wheel and fan case. We use spirally expanding cases, giving the blast wheel large clearance whether the fan is to be used to produce pressure or vacuum, regarding one as the mechanical analogue of the other, which greatly simplifies the consideration of fans, whereas the French, for producing pressure, use concentric close-fitting cases with small clearance, although for exhausting purposes they use the spirally expanding case, giving the wheel large clearance.

It has been stated that the air escapes from the tips of the blades of a blast wheel with the velocity with which the tips of the blades move, but in tests made with fans fitted with spirally expanding cases, giving large clearance to the blast wheel at all points and causing the formation of a rotating spiral vortex, it has been observed that the air moves at the same velocity as the tips of the blades only in cases of low peripheral speeds, and that as the peripheral speed increases, the increment of observed velocity above the tip velocity increases; that is, the observed velocities of the air issuing from the fan case increase as the revolutions of the wheel are increased in a greater ratio than the velocities of the tips of the blades, the air escaping from the fan case faster than the velocity due to the tip speed, the increment of velocity being found to be about one per cent. for each ounce increase of pressure. This augmentation of velocity comes from the fact that with a spirally expanding case there is formed between the wheel and its case a mass of revolving air, termed by Rankine a free vortex, the head at the circumference of this mass being the centrifugal head of the air in the wheel, plus the centrifugal force due to the revolving air, and clearly showing the advantage of a large case, permitting the formation of this vortex.

Much difference of opinion exists in regard to the proper proportioning of the wheels and case. It is claimed that the form of case greatly affects the efficiency of the blast wheel; while, upon the other hand, others claim that the form of case has little

or no effect upon the performance of any wheel. It is found that wheels of largely varying diameter and width at tips of blades can be used with economy in any given case. Where space is limited the size of the case is frequently decreased without perceptibly affecting the efficiency of the wheel, and in many cases where has been found a lack of capacity, the defect has been remedied by placing in the same case a wheel of larger diameter with greater width at tips of blades.

The effect produced by any wheel is largely independent of the size of the enclosing case, except that the clearance between wheel and shell may be so small as to interpose large frictional resistance and prevent the formation of a free vortex between wheel and case.

With spirally expanding cases it is necessary to give sufficient clearance between the tips of the blades and the throat to prevent roaring. When the blade tips are run too near the throat a very unpleasant noise is occasioned, and this noise can be done away with by increasing the clearance in any given case. The form of case greatly affects the noiseless action of the blower; the smaller the case the more noisy will be the wheel, and the larger the case the less the noise. It is not possible to make a fan perfectly noiseless, because rapid movement of air will produce sound; but any howling or roaring may be prevented by increasing the clearance between blast wheel and case.

The efficiency of a fan may be destroyed by a poorly proportioned system of piping. In designing a system of piping for conveying air, a proper allowance should always be made for friction by increasing the diameter of the pipe and by making all changes in direction in easy curves. Using the diameter of the blast pipe as a radius will give a curve productive of the best results. The enlargement of piping to compensate for frictional resistance should commence at the mouth of the fan, and at any point in the pipe the area should be at least equal to and preferably 20 per cent. greater than the aggregate area of all the branch pipes beyond through which air is being delivered. No mistake can be made by making the pipes too large, provided they are tight, and a serious mistake can be made by making them too

small. Enlargements and contractions in the pipe should be avoided in order to prevent loss of head.

It is well to remember in this connection that the loss of pressure due to frictional resistance of air flowing through pipes is directly as the length of the pipe and square of the velocity of the moving air and inversely as the diameter of the pipe. The fan should be located as near the work as possible in order to reduce the frictional loss to a minimum, and where a large amount of work is to be done, it is better to divide it among two or more fans than to put it on one fan, necessitating long distributing pipes and occasioning large loss by frictional resistance.

In the installation of a fan system for marine work particular care should be given to the arrangement of the blast pipe, for with a direct-connected engine the pressure of blast produced at the mouth of the fan is not large, and the pressure of blast can be so reduced from loss by frictional resistance as to render the fan practically inoperative. A fan should never be condemned until after a most careful investigation of the arrangement of the distributing pipe.

Table of pressures per square inch in ounces, from ¼ to 20 ounces.	Table of velocity in feet per minute of air (at 50° temperature Fahrenheit) escaping into open air through any shaped hole from any pipe or reservoir in which the air is compressed.	Table giving the number of cubic feet of air per minute (at 50° Fahrenheit) which may be discharged through a proper shaped mouth-piece, the diameter of which must be 1.366 inches, the area being 1.07 inches.	Table giving the actual H. P. contained in the blast discharged through the mouth-piece described in Table 3.	Table giving the number of cubic feet of air per minute that may be discharged with one H.P., no allowance being made for friction in the blast machine (whatever power that friction amounts to must be added). It makes no difference how the air is discharged, provided the pressure is steadily the same as given in the first table.
1.	2.	3.	4.	5.*†
¼	2,584.80	17.944	0.001224	14,662.76
½	3,657.60	25.400	0.003463	7,333.70
¾	4,482.00	31.124	0.005659	4,889 11
1	5,175.00	35.93	0.0098	3,666.62
2	7,338.24	50.96	0.0278	1,833.00
3	9,006.42	62.54	0.0512	1,222.30
4	10,421 58	72.37	0.0789	916.27
5	11,676.00	81.08	0.1106	733.39
6	12,817.08	89.01	0.1456	611.10
7	13,872.72	96.34	0.1839	523.81
8	14,861.16	103.20	0.2251	458.43
9	15,795.06	109.69	0.2692	407.42
10	16,683.51	115.86	0.3160	366.69
11	17,533.50	121.76	0.3652	333.40
12	18,350.34	127.43	0.4170	305.56
13	19,138.26	132.90	0.4712	282.05
14	19,900.68	138.20	0.5277	261.91
15	20,640.48	143.34	0.5864	244.44
16	21,360.00	148.33	0.6473	229.17
17	22,060.80	153.26	0.7103	215.77
18	22,745.40	157.96	0.7754	203.71
19	23,415.00	162.60	0.8426	192.98
20	24,070.80	167.16	0.9118	183.33

* See by this table how much more air can be discharged with one H.P. at low pressure than at high.

†This table shows the great advantage of large tuyers, large pipes, large blower and slow speed when the *nature of the work* will admit.

XXIX.

THEORY OF THE CENTRIFUGAL PUMP.

PART SECOND.

In the preceding paper the form of vane was assumed to be such as to give a constant angular velocity to the water, and the casing such as to give a constant radial velocity through the pump. This last condition is not always followed in practice, and parallel sides are sometimes used. Whether or no this is good practice can only be settled by experiment; but while it may be conceded that the velocity of the water may be changed from a high to a low one without loss of head, when this is done gradually, it is doubtful whether or no this can be accomplished with parallel sides, especially when the velocity of discharge is high.

These changes of velocity may be accomplished more gradually in the discharge nozzle by making the latter trumpet-shaped.

The form of vanes and degree of curvature has a great influence on the speed of rotation required to produce a given head. If we examine in detail the formulæ given for the total head, we will see that the first term gives that part of the total head depending on the difference of pressure produced between the entering and leaving edges of the vanes, and this difference of pressure is dependent upon the form of vane used. The other part of the total head depends upon the rotary velocity on leaving the tips of the vanes, and the proportions of the discharge conduit.

In order to fully determine algebraically all the elements entering into the design of the pump, the equation to the vane is necessary; for referring to equation (2) we have

$$\int \frac{dp}{\rho} = \int \frac{w^2 dr}{gr},$$

from which we see at once that the value of w in terms of r is necessary before we can determine the difference of pressure between the tips and the inner edges of the vanes.

In the preceding paper we showed that $w = \varphi r = ar - v \tan \theta$, where a = angular velocity of the wheel, v the radial velocity of the water (assumed as constant), and θ = the angle made by the vane with a radius drawn to the point in question. Substituting this value of w we obtain

$$\int \frac{dp}{\rho} = \int \frac{a^2 r^2 dr}{gr} - \int \frac{2arv \tan \theta dr}{gr} + \int \frac{v^2 \tan^2 \theta dr}{gr}.$$

In this equation the first term may be readily integrated, and will at once be recognized as the head produced by a straight vane. The second and third terms are dependent on the angle above mentioned. They may, however, be readily integrated approximately, giving a diagram of the vane to be used. We may regard these integrals as the integrals of an area, and, tabulating the value of $\frac{2av \tan \theta}{g}$ and $\frac{v^2 \tan^2 \theta}{gr}$ for various values of r, construct the area above mentioned, measure the same by means of a planimeter, or by Simpson's rules, and then obtain the value of the integral in question.

In illustration of the above we will assume a vane as shown on the plate. Let the wheel make 150 revolutions per minute. $r_1 = 2.5$ ft., $r_0 = 1$ ft. Then $a = 2\pi$ Rev. per sec. $= 15.7$. Radial velocity through wheel $= 600$ ft. per min. $= 10$ ft. per sec.

$$\frac{a^2 r_1^2}{2g}\left(1 - \frac{r_0^2}{r_1^2}\right) = \frac{15.7^2 \times 2.5^2}{64.4}\left(.84\right) = 20.22 \text{ ft.}$$

To determine the value of the second and third terms we draw circles dividing the vane radially into nine equal parts, and measure the angle θ at the point of intersection of these circles with the vanes. These values are tabulated, and in the present case the value of the areas found by means of the polygonal rules. Referring to Table 1 we obtain the value of the first integral $-$ 7.259, and Table 2 gives $+$ 1.162, leaving 6.09 feet less than would have been produced by a straight vane rotating with the same speed, and of the same dimensions. The vane at present under consideration has the advantage over the straight vane of avoiding shock at entrance.

The difference of head between the tips and the entering edges of the vanes will then be $20.227 - 6.097 = 14.130$ feet. Refer-

ring to Table 1, it will be observed that this area is to be subtracted from the head, and the less this quantity is made the greater the head produced. It will be observed, also, that by making the angle θ negative, this head may be increased; in other words, a blade bent forward will give the greatest head. Bending the blade too much forward would interfere with the radial flow of the water, but with great heads, and small radial velocity, it is advantageous to counteract the backward inclination of the entering edge of the vane by inclining it forward at the leaving edge. An inspection of the third term will show that this part of the head is positive, whether the vane is bent forward or backward. By the method given above, the head we may expect from any form of vane may be readily obtained.

TABLE 1.

θ	tan θ	$\frac{2ar}{g}$	Prod.	Mult.	
41°–34′	.886	9.75	8.638	½	4.319
39°–40′	.829	9.75	8.08	1	8.08
38°–30′	.795	9.75	7.74	1	7.74
36°–30′	.739	9.75	7.20	1	7.20
32°–40′	.641	9.75	6.14	1	6.14
26°–0′	.487	9.75	4.68	1	4.68
19°–0′	.344	9.75	3.35	1	3.35
10°–30′	.185	9.75	1.80	1	1.80
1°–30′	.026	9.75	.25	1	.25
0	0	9.75	0	½	0

6)43.559
— 7.259

TABLE 2.

θ	tan θ	tan^2 θ	$\frac{v^2}{gr}$	Prod.	Mult.	
41°–34′	.886	.78	3.1	2.41	½	1.20
39°–40′	.829	.69	2.65	1.82	1	1.82
38°–30′	.795	.63	2.32	1.46	1	1.46
36°–30′	.739	.55	2.06	1.13	1	1.13
32°–40′	.641	.41	1.86	.76	1	.76
26°–0′	.487	.23	1.65	.38	1	.38
19°	.344	.12	1.55	.18	1	.18
10°–30′	.185	.034	1.43	.042	1	.042
1°–30′	.026	.00067	1.32	.0008	1	.0008
0	0	0	1.24	0	½	0

6)6.9728
+ 1.162 ft.

With a spiral vane, as before stated, the angular velocity of the water is constant and different from that of the wheel. For comparison we will assume the wheel to rotate as before, making 150 revolutions per minute, while the water makes 120, so that $a = 2\pi \times 2 = 12.56$.

$$\frac{a^2 r_1^2}{2g}\left(1 - \frac{r_0^2}{r_1^2}\right) = \frac{12.56^2 \times 2.5^2}{64.4} \times .84 = 12.96.$$

The three forms of vanes then would stand to each other (for the same speed of rotation), so far as the head due to the difference of pressure alone is concerned, as follows:

Straight vane, ...	20.22
Compound curve (assumed), . . .	14.13
Spiral (assumed), ...	12.96

These ratios are considerably altered, however, when the other elements of the total head are considered.

For both the straight and the compound curved vane $\varphi = a = 15.7$ at the tip of the vane; hence we have

$$r_2 = r_1\sqrt{1 + \frac{\varphi^2 r_1^2}{v_1^2}} = 2.5\sqrt{1 + \frac{15.7^2 \times 2.5^2}{100}} = 10 \text{ ft. about.}$$

For the spiral vane $\varphi = 12.56$ and $r_2 = 2.5\sqrt{1 + \frac{12.56^2 \times 2.5^2}{100}}$ $= 8.1$ ft.

For the straight and the compound vane $v_2 = \frac{v_1 r_1}{r_2} = \frac{10 \times 2.5}{10}$ $= 2.5$; for the spiral vane $v_2 = \frac{10 \times 2.5}{8.1} = 3.08$.

The second part of the head will then be for the first two cases,

$$\left\{\frac{\varphi^2 r_1^2}{2g}\left(1 - \frac{r_1^2}{r_2^2}\right) + \frac{v_1^2 - v_2^2}{2g}\right\} = \frac{15.7^2 \times 2.5^2}{64.4}\left(1 - \frac{2.5^2}{10^2}\right) + \frac{10^2 - 2.5^2}{64.4} = 22.39 + 1.45 = 23.84.$$

And in case of the spiral vane,

$$\frac{12.56^2 \times 2.5^2}{64.4}\left(1 - \frac{2.5^2}{8.1^2}\right) + \frac{10^2 - 3.08^2}{64.4} = 13.79 + 1.7 = 15.49.$$

Adding these heads to those already obtained, we have

For the straight vane,	44.06
For compound,	37.97
For spiral,	28.45

The head due to the velocity of discharge is 1.55, and this must be subtracted from the above to obtain the height to which these vanes will respectively pump water with a velocity of 600 feet per minute.

From the above it will be seen that the compound curved vane is but slightly below the straight vane in pumping power for the same speed of rotation, and this difference might still be lessened by slightly inclining the tips forward on the leaving edge, while the shock at entrance may be wholly or partially avoided, thus saving power.

When the spiral vane is used, it should approach more and more to the radial form as the head is increased. A good rule is to make the vane normal to the relative path of the water through the wheel.

From the above example it will be observed that the diameter of the discharge casing is large, and from the figures given it will be seen that it might have been much reduced without greatly affecting the head, since $\left(1 - \frac{2.5^2}{10^2}\right) = 1 - .0625 = .9375$, nearly unity. The discharge conduit is often made as compact as possible, which is unfavorable to efficiency. However, it is not generally practicable to increase this diameter to the extent that theory would seem to call for, and when the discharge nozzle is made trumpet-shaped and joined to the casing at the proper angle, that diameter may be much reduced.

The power required to drive the pump, as well as the efficiency, for irregularly-shaped vanes, must be determined in a manner similar to that used above in determining the head.

Taking the example given with spiral vanes, we have the total head produced,

$$\frac{\varphi^2 r_1^2}{2g}\left\{\left(1 - \frac{r_0^2}{r_1^2}\right) + \left(1 - \frac{r_1^2}{r_2^2}\right)\right\} + \frac{v_1^2 - v_2^2}{2g} = 28.45 \text{ ft.}$$

Of this head 1.55 feet is necessary to produce the required

velocity of discharge, leaving 26.9 feet of effective head. The efficiency will be

$$\frac{\rho Q \times 26.9}{Fp} = \frac{26.9 \rho Q}{\frac{\rho Q \varphi^2 r_1^2}{g}\left(1 - \frac{r_0^2}{2r_1^2}\right)} = .95.$$

This efficiency is of course neglecting friction. Taking friction into account, we should expect, as in the case of turbines, about 75 to 80 % of this efficiency in practice, or about 71 to 76 %; and this efficiency will only be realized where the pump is pumping under the assumed conditions. If the speed remain the same, and the head against which the pump is working is lowered, the velocity of discharge will be greater, and the efficiency will consequently be less; on the other hand, if the head remain the same, and the speed of rotation is increased, the velocity of discharge will be greater and the efficiency will fall from this cause as well as from the fact that the discharge casing would not be adapted to this increased velocity of rotation.

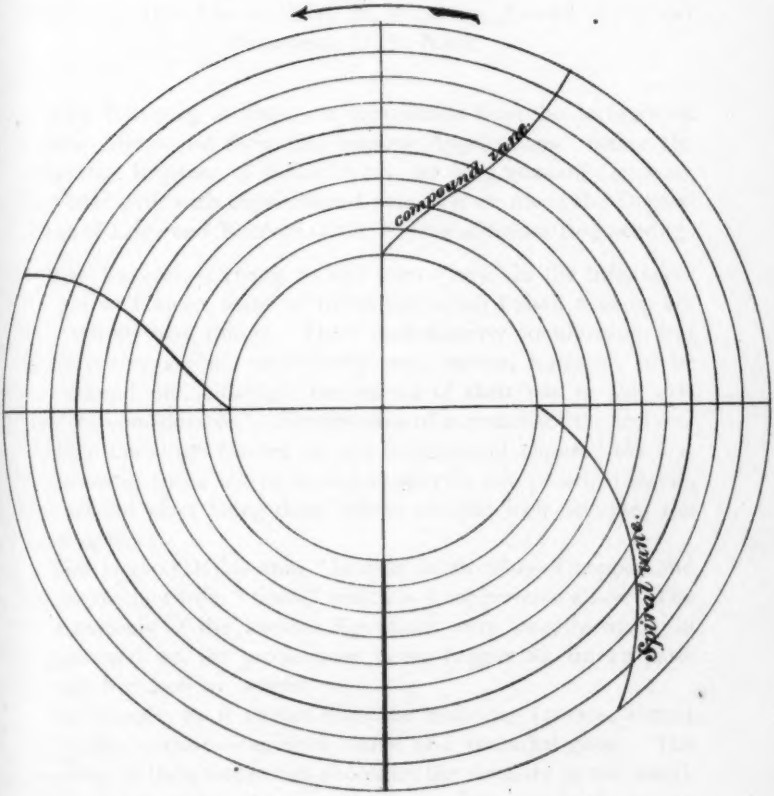
ERRATA TO PART ONE.

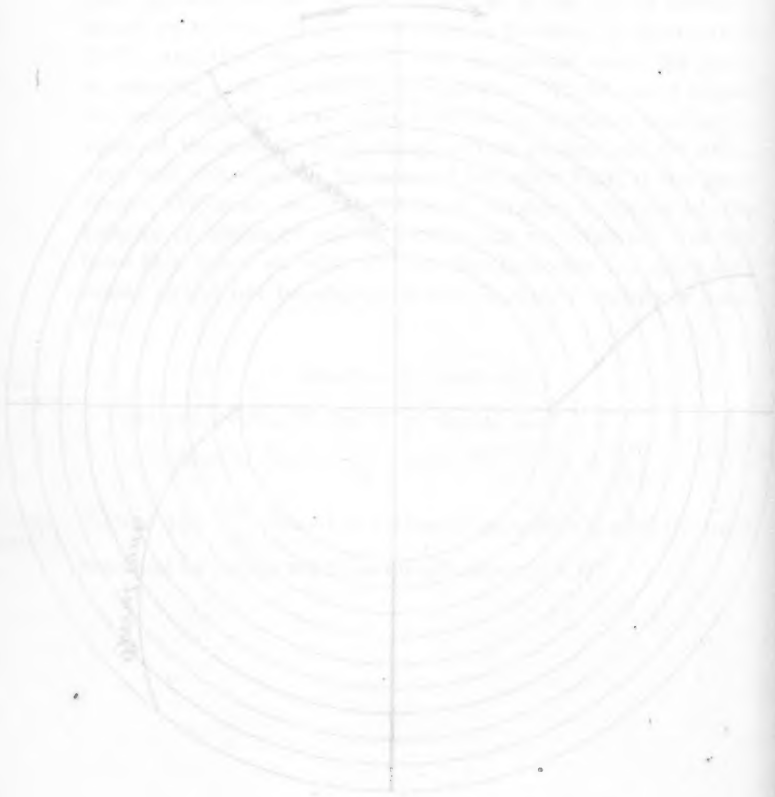
Page 338. Area of disc $\times$ p_1 should read

$$\text{Area of disc} \times (p_1 - p_0) = \frac{\rho \varphi^2 \pi r_1^4}{2g}\left(1 - \frac{r_0^2}{r_1^2}\right).$$

Page 341. $\frac{v_1^2}{2g}$ should be omitted from expression for the total head and be subtracted from the effective head H_e.

Fig. 5.





XXX.

SOME NEW ALLOYS CONSIDERED IN THEIR CHEMICAL AND PHYSICAL ASPECTS.

COMPILED AND ARRANGED BY H. WEBSTER, PASSED ASSISTANT ENGINEER, U. S. NAVY.

The following is mainly a compilation from the lectures on "New Alloys and their Engineering Applications," before the Franklin Institute of Philadelphia, by F. Lynwood Garrison, together with such experimental data as is on file in the Department of Logs and Records of the Bureau of Steam Engineering:

The lecturer, referring to the term "new" in the title, says: "I am well aware many of the alloys which I shall take up are not strictly new things. Their manufacture, composition, and peculiarities are not sufficiently well known, however, to be considered old, although the extent of their use in the arts may be considerable." For the sake of convenience the lecturer divided the alloys treated of into two general classes; the first class being those alloys having copper for the principal metal; the second class being those alloys wherein iron occupies the leading place.

The lecturer holds that "bronze" is an alloy of copper and tin, as distinct from "brass," which is a copper-zinc alloy. The bronze tools of the ancient Egyptians were usually made of copper and tin, the proportions being copper 88, tin 12 parts in one hundred by weight.

Pig copper, as it comes from the reducing furnace, almost invariably contains cupreous oxide and occluded gases. The presence of these impurities decreases the ductility of the metal, and their removal produces an astonishing increase in the tensile strength and general ductility.

In this simple fact, says Mr. Garrison, lies the secret of the excellent results obtained with the improved copper alloys

known as aluminum bronze, phosphor bronze, manganese bronze, deoxidized copper, tempered copper, &c.

Aluminum, phosphorus, manganese, arsenic, silicon, and possibly iron, when added in various ways to copper, act on these traces of cupreous oxide dissolved in the copper, deoxidizing the same, and thus improving and toughening the copper.

Aluminum bronze is the first alloy treated of by the lecturer, and its composition and properties are deduced from the analyses of Mr. James S. De Benneville, of Dr. F. A. Genth's laboratory.

The specimens analyzed were from the material used for firing pins by the Colt Fire Arms Company, of Hartford, Conn.

ALUMINUM BRONZE.

	I.	II.
Copper,	91.260	91.260
Aluminum,	7.410	7.410
Silicon,	0.930	0.930
Iron,	0.220	0.301
Lead,	0.570	0.470
Phosphorous,	0.056	0.025
Arsenic,	0 042	trace
	100.488	100.396

The iron and silicon were probably present as impurities in the aluminum.

The melting point of this bronze varies somewhat with the amount of aluminum contained, the higher grades melting at a lower point than the lower grades. This bronze shrinks more than ordinary brass, and as it solidifies rapidly it is necessary to pour quickly and to have the feeders large.

Aluminum bronze usually gives the best results when the proportions of its constituents are above ninety per cent. copper and ten per cent. aluminum. With small bars cast in sand the best physical test results were: elastic limit 70,000 pounds per square inch, tensile strength 95,000 pounds per square inch, with an elongation of about ten per cent. With rolled bars better results have been obtained. The modulus of elasticity of

aluminum bronze is about 18,000,000.' Specific gravity, cast, 7.56, rolled, 7.89.

It is worthy of note, in connection with the foregoing, that these are the best results, and that the average run of ten per cent. aluminum bronze will give results somewhat lower than these, and analysis will probably show less than ten per cent. of aluminum.

The addition of aluminum bronze to ordinary brass (copper-tin-zinc alloy) improves the latter metal in strength, toughness, and resistance to corrosion.

Tests have been made with aluminum brass showing a tensile strength as high as 95,000 pounds per square inch. This alloy is claimed to possess the ability to withstand a high temperature without loss of strength.

The next alloy is that of manganese with copper, the best known examples of which are the various manganese bronzes of Mr. P. M. Parsons, of the Manganese Bronze Company, Deptford, England.

This company make five grades of manganese bronze, No. 1 of these being used wherever strength is required, Nos. 4 and 5 being intended for anti-friction purposes. No. 1 is the only grade which cannot be cast in sand, chill moulds being employed. No. 2 is the grade usually employed for propeller blades, hydraulic rams, &c. The following analysis of a specimen of Parson's No. 2 manganese bronze was made, as in the preceding example, by Mr. De Benneville, and the specimen was a chip from the propeller of Mr. W. K. Vanderbilt's yacht *Alva*.

PARSON'S MANGANESE BRONZE.

No. 2.

Copper,	88.644
Zinc,	1.570
Tin,	8.700
Iron,	0.720
Lead,	0.295
Phosphorus,	trace.
	99.929

It will be observed there is no manganese present, and the percentage of zinc is very low.

Manganese bronze has been used extensively, both in this country and in England, for casting propeller blades. Some of the best results yet obtained have been with this material made by Messrs. B. H. Cramp & Co., of Philadelphia. The alloy produced by this firm, when cast in sand, has an average elastic limit of 30,000 pounds per square inch, a tensile strength of 60,000 pounds per square inch, and an elongation of from eight to ten per cent. When rolled, the elastic limit is increased to 80,000 pounds per square inch, and the tensile strength to 106,000 pounds per square inch, with an elongation of fifteen per cent.

These tests go to show that in this alloy as produced under the careful supervision of the foundry carried on by this company, we have one of the strongest "new" metals yet experimented with.

The manganese bronze manufactured by B. H. Cramp & Co. varies materially in its composition from the metal furnished by the Manganese Bronze Company of Deptford, England, operating the patents of Mr. P. M. Parsons, but owes its superiority, in a great measure, to the care exercised in manipulating the constituents of the alloy, extreme pains being taken to secure purity of materials, exactness in weights, and regularity of time in melting. A marked difference, however, between the metal of the Parsons Company and that of the Cramp Company lies in the fact that whereas the former metal fails to show any manganese under analysis, the bronze manufactured by the Cramp Company shows not less than five per cent. of manganese under careful analysis.

Parson's metal as delivered from the rolls possesses a tensile strength of 67,200 pounds per square inch, an elastic limit of 50,000 pounds per square inch, and an elongation of from 23 to 25 per cent. By the process of cold rolling the tensile strength is increased to a maximum of 90,000 pounds per square inch, an elastic limit of from 67,200 to 76,000 pounds per square inch, and an elongation of 10 per cent. The annealing process changes its tensile strength very slightly, but the elastic limit

is reduced one-half, and the elongation increased to 30 or 35 per cent.

Parson's No. 2 manganese bronze, cast in sand, averages about as follows:- Elastic limit, 35,000 to 43,000 pounds per square inch. Tensile strength, 63,000 to 75,000 pounds per square inch. Elongation, from 13 to 22 per cent. in five inches.

Under the general title of "Copper-Zinc-Iron Alloys," the well-known delta metal is considered. This alloy was formerly known as sterro-metal, introduced many years ago by Baron Rosthorn, of Vienna, and was mentioned under this title as far back as 1864.

Delta metal, as such, was patented by Dick, in England, in 1882, and according to the specification may or may not contain tin.

The general composition of this alloy is—

Copper, ...	60 parts by weight.
Zinc, ...	34 to 44 parts.
Iron, ...	2 to 4 parts.
Tin, ...	1 to 2 parts.

The following will indicate the variable nature of this singular alloy, and shows upon what a changing base the claim for protection by patent is founded:

	I.	II.
Copper, .	50 to 65 per cent.;	40 to 98 per cent.
Zinc, .	30 to 49.9 per cent.;	1.8 to 45 per cent.
Iron, .	1 to 5 per cent.;	1 to 5 per cent.
Tin, ...		1 to 10 per cent.

The iron in these alloys seems to have the property of increasing their strength to an unusual degree.

In the manufacture of delta metal the iron is previously alloyed with zinc in known and definite proportions. When ordinary wrought iron is introduced into molten zinc this latter metal readily dissolves or absorbs the iron, and will take it up to the extent of five per cent. or more. By adding this zinc-iron alloy to a definite quantity of copper, it is possible to produce

an alloy containing a definite quantity of iron up to five per cent. "Aich's metal" is practically the same.

When cast in sand delta metal has a tensile strength of 45,000 pounds per square inch with 10 per cent. of elongation. When rolled the tensile strength rises to from 60,000 to 75,000 pounds per square inch, the elongation increasing also to as high as 17 per cent. The patentee, Mr. Dick, claims that by his process the iron is chemically combined in the bronze.

Delta metal produces sound castings of close grain, and possesses great strength and toughness.

The copper-zinc-tin alloy patented by Passed Assistant Engineer John A. Tobin, U. S. N., was analyzed by Mr. De Benneville with the following results:

Copper,	61.203
Zinc,	37.440
Tin,	0.906
Iron,	0.180
Lead,	0.359
	100.088

This alloy is therefore practically a brass, and owes no little of its strength to the amount of iron contained. In many respects Tobin bronze is another form of sterro-metal, or delta metal, with the addition of a small amount of lead, which tends to increase its softness and ductility.

The following analyses of Tobin bronze were made by Dr. Chas. B. Dudley, of Altoona, Pa.:

	Pig metal, per cent.	Test bar (rolled), per cent.
Copper,	59.00	61.20
Zinc,	38.40	37.14
Tin,	2.16	0.90
Iron,	0.11	0.18
Lead,	0.31	0.35
	99.98	99.77

Tobin bronze is manufactured solely by the Ansonia Brass and

Copper Company, of Ansonia, Conn., is claimed to have a tensile strength of 79,600 pounds per square inch, an elastic limit of 54,257 pounds per square inch and from 12 to 17 per cent. of elongation, the specimens being from best rolled one-inch bars.

From some recent tests made on behalf of the United States Navy Department, Bureau of Steam Engineering, at the New York Navy Yard, the following results were obtained with forged Tobin bronze valve stems:

These forgings exhibited no defects, and were accepted by the Government inspector. On breaking, the metal was found to be of fine homogeneous grain and perfectly sound. The maximum tensile strength developed was 57,000 pounds per square inch of original section of specimen.

Under the head of copper-phosphorus alloys, three compositions are enumerated. Phosphor bronze, deoxidized bronze, and tempered copper, this latter the production of the Eureka Tempered Copper Company.

The discovery of the great improvements in strength and ductility of copper by adding a small percentage of phosphorus to that metal was made in 1868 by Montefiore & Künzel, of Liège, Belgium. Phosphor bronze, in common with several of the alloys under discussion, has a composition varying with the uses to which it is to be applied. In a specimen analyzed by Dr. Chas. B. Dudley, the following components were found:

Copper,	79.70
Tin,	10.00
Lead,	9.50
Phosphorus,	0.80
	100.00

In Brannt's work on metallic alloys the following analyses are given for phosphor bronze:

	I.	II.	III.
Copper,	90.34	90.86	94.71
Tin,	8.90	8.56	4.39
Phosphorus,	0.76	0.196	0.053
	100.00	99.616	99.153

It will be observed that the quantity of phosphorus present is quite small. This, however, is only what might be expected if the supposed role played by phosphorus in the alloy is the correct one. The most general method of introducing the phosphorus required for the alloy is by the addition of phosphor tin, which is made by heating three parts of anhydrous phosphoric acid with one part of carbon and six parts of tin. The resulting alloy has a silvery white crystalline appearance, and, according to Pelletier, has the composition $Sn_3 P_2$.

From some experiments made by Kirkaldy, the eminent Scotch engineer and physicist, the following results were obtained with phosphor bronze.

The specimens tested were cast, and were furnished by the Phosphor Bronze Company, London:

Elastic limit. Pounds per sq. inch.	Tensile strength. Pounds per sq. inch.	Elongation. Per cent.
23,800	52,625	8.40
24,700	46,100	1.50
16,100	44,448	33.40

In the form of wire, as drawn, without subsequent treatment, phosphor bronze has shown a strength as high as 159,515 pounds per square inch.

It will be noted that the analysis by Dr. Dudley, before quoted, shows about 9.5 per cent. of lead, and Dr. Dudley states that this analysis will not vary more than one or two per cent. in the hundreds of thousands of pounds which have been received and used by the Pennsylvania Railroad Company.

Phosphor bronze is largely employed not only as a metal for bearings, but for pump cylinders, hydraulic presses, piston rings, propeller blades, wire, bells, and gunpowder machinery.

The metal produced by the Deoxidized Metal Company, of Bridgeport, Conn., bears considerable resemblance to phosphor bronze, and also to delta metal, containing, as it does, both iron and zinc. The following analysis was made by Mr. De Benneville in Dr. Genth's laboratory.

DEOXIDIZED BRONZE.

Copper,	82.67
Tin,	12.40
Zinc,	3.23
Lead,	2.14
Iron,	0.10
Silver,	0.07
Phosphorus,	0.005
	100.615

It seems probable that some deoxidizing flux containing phosphorus, similar to that used in the manufacture of phosphor bronze, is employed in the manufacture of this alloy.

Deoxidized bronze is used for wood-pulp digesters, it being found to resist the action of sodium hyposulphite and sulphurous acid remarkably well.

Deoxidized bronze wire possesses a tensile strength of about 150,000 pounds per square inch, and the deoxidized copper wire made by this company has a tensile strength of 70,000 pounds per square inch.

Eureka Tempered Copper is the name applied to the product of the company established at Northeast, Pa., but it would seem that the term "tempered" is a misapplication of the word, as the material is not in any sense tempered or unusually hardened, but is practically pure copper.

The following comparisons of Lake Superior pig copper, from which castings were made, and the "tempered" castings themselves, show the composition of the metal to have been but slightly affected by the so-called "tempering" process.

The presence of minute quantities of arsenic and phosphorus in the "tempered" product indicate their use, probably as a deoxidizing flux, the chemical reactions taking place being similar to those occurring in the manufacture of all deoxidized alloys.

These analyses were made by Dr. F. A. Genth, of Philadelphia:

	Pig copper.	Not tempered.	Tempered.
Copper,	99.890	99.930	99.981
Silver,	0.096	0.026	0.025
Tin,	trace.	none.	none.
Lead,	trace.	——	——
Iron,	0.056	0.082	0.088
Zinc,	——	none.	none.
Aluminum,	——	none.	none.
Arsenic,	——	0.046	0.042
Phosphorus,	——	0.017	0.018
	100.042	100.101	100.154

The treatment does not appear to affect the physical properties to any extent, in fact it would seem that in one or two respects the untreated pig copper was better than the treated metal.

The following are some of the results of tests on specimens of the products of the Eureka Tempered Copper Company:

The "untempered" cast material gave a tensile strength of 24,410 pounds per square inch, an elastic limit of 10,080 pounds per square inch, and an elongation of 22.5 per cent.

The "tempered" cast metal gave a tensile strength of 25,800 pounds per square inch, an elastic limit of 11,460 pounds per square inch, and an elongation of 18 per cent.

The "tempered" rolled metal gave a tensile strength of 61,290 pounds per square inch, an elastic limit of 27,000 pounds per square inch, and an elongation of 4.3 per cent. in six inches.

A comparative compression test of Lake Superior pig copper, untreated by any process and "untempered," and a specimen of Eureka "tempered" copper, gave for the first named, with a compressive strain of 50,000 pounds per square inch, a compression of 21 per cent., and for the second or "tempered" specimen, with a compressive strain of 51,600 pounds per square inch, a compression of not less than 27.9 per cent. From which it appears that untreated pig copper stood the strain of compression better than the treated material.

Silicon bronze was invented in 1881 by M. Wieller, of Angouleme, in the course of experiments with phosphor bronze for

telegraphic use. The action of the silicon on the copper is similar to that of phosphorus; it acts as a deoxidizer, and, the silica formed being an acid, is a valuable flux for any metallic oxides remaining unreduced.

The silicon-copper compound, by means of which silicon bronze is produced, is made by melting, in a graphite crucible, a certain amount of copper with a mixture of fluor-silicate of potassium, glass, chloride of soda, carbonate of soda, and chloride of calcium. It is claimed in the patent specifications that the silicon and sodium in this mixture absorb all the oxides present in the mass.

Wire made of this alloy is said to have a strength equal to that of phosphor-bronze, but with a much higher electric conductivity.

Silicon bronze is used principally in the production of electric wire, its greater strength and lightness rendering it particularly valuable for this purpose.

The fact that moderately high temperatures affect the strength of the commonly used alloys of copper is a question of no little practical importance, and all experimental data goes to show that the variation in strength bears but little relation to the increments of temperature after certain moderate heat has been applied.

As far back as 1877, Professor W. C. Unwin made some experiments for the Board of Admiralty on the effect of temperature on the strength of certain copper alloys, Muntz metal and phosphor bronze. Some later experiments by the same authority give the following results:

Metal.	Temperature Fahr.	Tenacity in lbs. per sq. inch.	Metal.	Temperature Fahr.	Tenacity in lbs. per sq. inch.
Yellow brass.	Atmos.	53,961.6	Gun metal.	210°	26,119.0
"	258°	50,265.6	"	380°	27,460.0
"	400°	47,553.2	"	406°	24,777.0
"	500°	41,059.2	"	440°	27,550.0
"	602°	35,526.4	"	500°	17,556.0
"	640°	32,457.6	"	600°	17,560.0
			"	615°	10,797.0
Delta metal rolled	Atmos.	69,808.4	Cast brass.	Atmos.	27,895 0
"	260°	62,992.0	"	350°	26,500.0
"	400°	59,539.2	"	450°	23,300.0
"	500°	53,379.2	"	500°	17,225.0
"	570°	43,376.8	"	550°	17,199.0
"	650°	35,929 6	"	645°	7,235.0
Muntz metal.	Atmos.	55,280.0	Phosphor	Atmos.	35,970.0
"	300°	51,150.0	Bronze cast.	270°	31,720.0
"	400°	44,695.0	"	350°	28,760.0
"	500°	42,139.0	"	430°	27,800.0
"	600°	37,390.0	"	500°	24,865.0
"	650°	38,420.0	"	600°	18,297.0

IRON ALLOYS.

When metallic iron is alloyed with various proportions of carbon, the results are the ordinary varieties of iron and steel, the purest form being wrought iron, containing not more than .015 per cent. of carbon; the next being steel, with from .015 to 1.5 per cent. of carbon; and, lastly, cast iron, with upward of 1.5 per cent.

There are other elements nearly always present which to a greater or less degree affect the properties of the metal.

It is the purpose of this article to confine attention to a class of alloys which are now attracting much attention and may be considered as essentially new. Reference is made to the combinations of iron with such elements as aluminum, chromium, copper, manganese, nickel, silicon and tungsten, these alloys being called steels, prefixing the name of the predominating and particular element present.

One of the most serious disadvantages connected with the use of mild steel in shipbuilding is its greater liability to corrosion in sea water than iron. This is probably due to the generally

greater purity of mild steel. In several of the new steel alloys, more particularly nickel steel, this difficulty is overcome, as in them we have a particularly homogeneous material of even greater strength than mild steel, and much less corrodible in sea water. Some of these alloys seem to be particularly well adapted for rivets. This is an important point. The London Steel Company found, as the results of many experiments, that the limit of strength of steel best suited for shipbuilding was not determined so much by the quality of the steel plates employed as by the quality of the steel rivets used.

ALUMINUM STEEL.

The use of ferro-aluminum and aluminum steel in the arts is quite new, and it can hardly be said we know much of their properties and peculiarities. From recent experiments made by Mr. James Riley, of Scotland, it appears that the addition of aluminum slightly increases the breaking strain, and proportionally the elastic limit, in rolled and cast steel according to the amount added, not exceeding one per cent., further addition doing no good. It also improves the extension in rolled steel up to 0.5 per cent., a larger amount reducing extensibility. Its addition is useful, where castings of comparatively thin metal are to be run, because of a consequent greater fluidity of the steel. On the other hand contraction cracks are more liable to occur unless the temperature of the castings is kept low.

In February, 1886, at the meeting of the American Institute of Mining Engineers in Pittsburgh, Mr. Petter Ostberg, of Stockholm, Sweden, exhibited specimens of exceptionally good malleable iron castings (mitis metal), which were claimed to be produced from wrought iron or steel. His statement was: "We heat the wrought iron just to melting, but not more, and as soon as molten we add from 0.05 to 0.1 per cent. aluminum, thereby producing a sudden lowering of the melting point."

Recent investigations have shown this statement to be open to question, it being a well-known fact that ordinary steel sometimes contains small quantities of aluminum without having its melting point affected thereby.

Mr. Howe states that the most careful analysis has failed in detecting aluminum in these mitis castings.

Analyses of mitis metal by Dr. Edward Riley show, as an average of seven samples:

Carbon,	0.099
Silicon,	0.100
Phosphorus,	0.153
Manganese,	0.022

No aluminum was found. It may be remarked that the above composition is about that of ordinary wrought iron with the slag removed. Omitting from consideration the action of aluminum on cast iron, it is found, according to Keep's researches, that the general effect of aluminum (or ferro-aluminum) when added to non-carburetted iron just before turning, is quieting. While the addition of ferro-aluminum makes non-carburetted iron more fluid, it does not seem to act in the same way on steel with over 0.25 per cent. of carbon.

Mr. Keep observes that, in whatever way the end is accomplished, aluminum seems to permit good castings to be made from metal which could not otherwise be used.

He also states that the presence of 0.20 per cent. of aluminum makes the grain of ingot iron (mild steel) close and uniform, removes blowholes, increases the ultimate strength fifty per cent., but diminishes the tendency to flow. Power to resist shock is almost doubled.

On the other hand Mr. R. A. Hadfield casts doubt on the conclusions of the preceding authority, considering it an open question whether aluminum increases the fluidity of properly made steel or the reverse.

Furthermore it does not appear from Mr. Hadfield's experiments that the addition of aluminum improves the soundness of castings any more than does silicon, manganese, or silicon combined with manganese. Hadfield considers that the action of aluminum may be classed with that of silicon, sulphur, phosphorus, arsenic, and copper, as giving no increase of hardness to iron, in contradistinction to carbon, manganese, chromium, tungsten and nickel.

Water-quenching produces no effect upon either forged or cast aluminum steel.

Here we have another addition to the list of elements which, alloyed with iron, do not produce "water-quenched hardness," or the property of tempering. In fact, notwithstanding that manganese, chromium, tungsten, and nickel do impart a certain and considerable kind of hardness to iron, either in the cast or forged state, carbon appears to be the only element which confers upon iron the property of becoming hard by water-quenching. In steel made with the hardening metals just mentioned, if the carbon present be low, no matter at what heat they are water-quenched, the characteristic of carbon steel, such as scratching glass or forming the edge of a cutting tool, is wanting.

The tensile strength of iron is not increased so much by the addition of aluminum as of silicon.

It has been found that these special steels, or alloys, such as silicon, manganese, or chromium steel, have but slight, if any, welding properties.

CHROME STEEL.

Chromium increases the hardness of iron; perhaps, also, the tensile strength and elastic limit.

Ferro-chrome is made by strongly heating the mixed oxides of iron and chromium in charcoal-lined crucibles, adding powdered charcoal, if the oxide of chromium is in excess, and fluxes to scorify the earthy matter and prevent oxidation. This is the method employed at Brooklyn, N. Y., and at the Unieux Works, in France, where chrome steel has been produced in large quantities for a number of years.

Chrome steel is made from ferro-chrome by melting it with wrought iron or steel in graphite crucibles.

The physical properties of chrome steel vary greatly, according to the relative proportions of the other elements present in the alloy. In general it may be considered that chromium has a tendency to harden steel, and in most cases it increases somewhat the tensile strength and elastic limit. Its ductility, however, when subjected to elongation, is not necessarily increased.

The Holtzer Company (Unieux, France,) have, however, for

a number of years, been making remarkably hard and tough projectiles of chrome steel. It is stated by Howe that in many specimens of so-called chrome steel analyzed by experienced chemists, no chromium at all, or but the merest trace, has been found.

COPPER STEEL.

A few years ago M. Henri Schneider, of Creusot, France, took out patents for the manufacture of alloys of iron and copper and steel and copper. In the patent specifications it is stated that the alloy of cast iron and copper can be made in a crucible, cupola, or open-hearth furnace. The furnace is charged with copper scrap and cast iron mixed between layers of coke; or if a cupreous coke be employed, then the cast iron is laid in alternate layers with it, and a layer of anthracite is preferably laid over the whole. The alloy thus formed contains, generally, from five to twenty per cent. of copper, according to the purpose for which it is to be employed, and it is remarkable for its great strength, tenacity and malleability.

For the production of copper steel this alloy is charged into the bed of a furnace, with the ordinary ingredients used in the manufacture of steel, preferably under a layer of anthracite, to prevent oxidation.

Mr. James Riley, of Glasgow, finds that if an alloy of steel and copper be examined under the microscope, it is found that the copper is not alloyed at all, but is disseminated throughout the piece. Copper steel alloys are almost too new to determine for what particular purposes they would be most useful, although it is claimed that they are valuable for making ordnance, armor plate, rifle barrels, and projectiles. In view of the remarkable elastic limit of copper steel, while maintaining at the same time a very considerable elongation, its use may become very extensive in the arts.

The following tests of some copper steels exhibited at the Paris Exhibition of 1889, by the Unieux Steel Works, will illustrate the preceding paragraph. The specimens contained from three to four per cent. of copper:

No.	Elastic limit. lbs. per. sq. inch.	Tensile strength. lbs. per sq. inch.	Elongation. per cent.
1,	43,563	77,761	22.5
2,	99,330	109,972	17.5
3,	94,789	113,236	13.0
4,	65,274	83,721	18.6
5,	141,900	173,543	6.2
6,	121,324	149,420	11.0
7,	94,789	121,324	14.5
8,	———	156,090	2.0

1 and 4. Natural state.

3, 5 and 8. Oil hardened and tempered.

2, 6 and 7. Oil hardened and annealed.

MANGANESE STEEL.

The effect of small amounts of manganese on the strength and ductility of steel appears to be slight, in fact it diminishes both these properties if its amount exceeds 2.5 per cent., conferring, however, remarkable hardness. This effect reaches a maximum when the manganese is about five or six per cent. With a further increase the strength and toughness increase, while the hardness somewhat diminishes.

The maximum of both strength and toughness is probably reached with about fifteen per cent. of manganese, the hardness still remaining so great that the metal can scarcely be machined.

It is stated on good authority that the best composition is that of fourteen per cent. of manganese, and not over one per cent. of carbon. Such a metal is very fluid while molten, solidifies rapidly and with great contraction, does not form blow-holes, but pipes deeply, is forgeable, but welds poorly.

The chief obstacle to the commercial introduction of manganese steel is its hardness. The machining of some of the grades of this material is almost impossible by the ordinary methods.

Manganese steel not being so liable to honeycombing as ordinary steel, the addition of silicon, to obviate unsoundness, is not necessary. Its great fluidity enables it to be run in thin sections.

In the cast state manganese steel is extremely brittle and hard,

but when forged, a remarkable increase in strength takes place, the tensile strength averaging about 56,000 pounds per square inch, with three per cent. elongation. Howe states that, "under stress manganese steel acts very differently from wrought iron and carbon steel. With 125,000 pounds of ultimate tensile strength it may begin taking serious permanent set under stress of 35,000 pounds per square inch, so that in this respect it is little better than common soft steel with 60,000 pounds tensile strength."

The chemical reactions taking place in the manufacture of manganese steel are probably very much the same as those taking place in the manufacture of manganese bronze. The dissolved oxide of iron present in the steel is deoxidized by the manganese in the same way as the dissolved cupreous oxide present in the metallic copper is deoxidized in making manganese bronze.

The presence of manganese in solidified steel appears to hinder its oxidation in heating and forging, and also is said to counteract "red shortness," caused by the presence of sulphur.

In analyzing manganese steel no trace of graphite has been found, all the carbon present being in the combined form.

NICKEL STEEL.

Mr. James Riley, of Glasgow, was the first to call attention to the subject of nickel and steel alloys, in 1889. In the same year Mr. H. Schneider, of Creusot, France, took out patents for the manufacture of alloys of cast iron and nickel, and steel and nickel, respectively.

What is known as "Marbeau's nickelo-spiegel" is made by a patented process which consists in the reduction of the ores of nickel, iron and manganese at the same time and in one operation. The following proportions are stated as affording good results:

Nickel ore, containing 10 per cent. of nickel,	2 tons.
Manganese ore, containing 10 per cent. of manganese and 40 per cent. of iron,	1 ton.
Iron ore, containing 50 per cent. of iron,	12 cwt.

An alloy of nickel and iron having been thus obtained, there

appears to be but little difficulty in working this nickel iron into a nickel steel by means of an open-hearth furnace.

If the charge is properly worked, nearly all of the nickel will be found in the steel, almost none being lost in the slag. With steel containing 5 per cent. of nickel, and rolled but not annealed, an elastic limit of 69,664 pounds per square inch has been obtained by Mr. Riley, the tensile strength being 116,480 pounds per square inch, with 14 per cent. elongation in eight inches. Riley states that the whole series of nickel steels, containing up to 50 per cent. nickel, takes a good polish and finish. Steels rich in nickel are practically non-corrodible. Steel containing 1 per cent. nickel welds fairly well, but this quality deteriorates with each addition of nickel.

Mr. J. F. Hall, of Sheffield, England, states that he has made nickel steel with a tensile strength of 217,280 pounds per square inch and over 7 per cent. elongation.

Howe states that the hardness of this alloy depends on the proportion of nickel and carbon jointly, nickel up to a certain percentage increasing the hardness, beyond this lessening it. Thus steel with 2 per cent. nickel and 9 per cent. carbon cannot be machined.

The presence of manganese in nickel steel is most important, as it appears that without the aid of manganese in proper proportions the conditions of treatment would not be successful. Nickel steel does not seem to be nearly so liable to corrosion in salt water as the ordinary and purer grades of steel. It was remarked above that the corrosive action of salt water is much greater on ordinary steel than on the more impure qualities of iron. Nickel steel would, therefore, seem to be an advance, as it appears to combine the qualities of mild steel for structural purposes with a greater limit of elasticity and less liability to corrosion than either ordinary steel or iron.

SILICON STEEL.

Up to a few years ago it was a common belief that silicon in any considerable amount was decidedly injurious to the strength and ductility of steel. It is now found that considerable quanti-

ties of silicon may be combined with steel to advantage, provided the quantity of carbon be very small. The role played by silicon in cast iron and steel is an interesting one, and from the investigations of Keep and Turner, iron which would form porous and brittle castings is made free from honeycombs and given greater strength by the addition of comparatively small amounts of silicon.

Investigation seems to show beyond a doubt that the addition of silicon to carbon steel changes the combined carbon to the graphitic state.

In the Bessemer process of steel manufacture silicon is necessary for the production of a good metal, the amount being about 1.5 per cent. The effect of silicon in this process is to allow of the temperature being kept higher than is otherwise attainable. Silicon steel is made from a rich, silicious pig-iron, known as "ferro-silicon," in much the same way as manganese steel is produced from ferro-manganese. Ferro-silicon is practically a pig iron containing from seven to 14 per cent. of silicon, and of a very high grade in other respects.

Silicon has a remarkable affinity for iron, recent investigations having shown that iron will combine with seven or eight times more silicon than carbon. The loss of silicon in remelting silicon irons is trifling. The best results have been obtained with about two per cent. of silicon and about 0.19 per cent. of carbon.

Silicon resembles manganese in that steel containing it cannot be tempered.

One of the effects of silicon on steel is probably similar to that of manganese. The silicon reduces part of the oxide of iron present in the steel, forming silica, and this, combining with the remainder of the oxide of iron, forms a silicate which rises to the top and runs off as a flux.

TUNGSTEN STEEL.

Although the metal tungsten is infusible, it readily unites with iron, forming an alloy known as "ferro-tungsten." In this form it is used in the preparation of tungsten steel.

The process of making tungsten steel is similar to the manu-

facture of ordinary steel, in which the manganese oxide and the tungstic oxide change places. The steel is first melted, not carbonized, and ferro-tungsten of known composition is added to the same. The general effect of tungsten on steel is to render it intensely hard and brittle. It is very difficult to forge, and cannot be welded when the amount of tungsten exceeds ten per cent. It is, therefore, unfit for structural purposes, and also for tools subject to shock, such as rock drills, chisels, hammers, &c.

Tungsten steel, after a few re-heatings, becomes oxidized and loses its special properties.

In France the St. Chamond Company use tungsten steel for springs, claiming it to possess a carrying power about one-third greater than the best carbon spring steel.

Tungsten steel is not, in the absence of carbon, strengthened or increased in hardness by tempering.

The higher grades of this steel cannot be cut by the file, and as none of it can be tempered, it must be shaped at one forging, and then ground to the form desired.

Tungsten steel is obviously especially useful for standard gauges, tools, &c.

XXXI.

ELECTRIC LIGHTING OF SHIPS.

Under the direction of Lloyd's Register, Superintending Engineers Milton and Allison some time ago made a report on electric lighting of ships, and, based on their observations, worked out a system of regulations governing such installations, the principal portions of which are given below.

The authors have consulted not only the manufacturers and fitters of electric lighting plants, but also those in charge of them on board of ships. They have studied the dangers attending the use of electric light and the means to avert them.

These dangers are twofold: 1, the disturbing influence exercised by the field magnets and the current traversing the wires on the ship's compass; 2, the danger of fire incidental to a faulty installation.

As regards the effect of the field magnets of the dynamo, either directly or by magnetism induced in surrounding masses of iron, this is capable of decrease by multiplying the magnet poles, but can never be entirely overcome. The only remedy is to place the dynamo as far away from the compass as possible.

The effect on the compass of the current passing over the wires may, however, be made practically nil. With the double-wire system the effects of the current going and coming neutralize one another—at any rate, if the wires are not too far apart and not too near the compass. With the single-wire system the effect of the current on the compass may be very great, unless the wire is sufficiently far removed. For currents that pass of necessity near the compass double wires must be arranged. To meet the dangers growing out of these disturbances, it is recommended to determine the compass errors in the usual way first, then to put the ship on the different corrected courses and note the error induced by starting up the dynamo. The disturbances due to the

dynamo itself may be separated from those due to the currents in the wires by cutting out those passing in the neighborhood of the compass and noting the error.

In order to understand the dangers attending the excessive heating of any portion of the conducting wire, it is necessary to bear in mind the laws governing this heating:—the heat generated is proportional to the resistance of the medium, to the square of the strength of the current, and to the time during which the current flows. The rise in temperature is dependent not only on the magnitude of the resistance, but also on the kind of resistance offered. A short stretch of thin wire may have the same resistance as a great length of a thicker wire; the quantity of heat generated in equal times by equal currents will then be the same, but the rise in temperature in the short piece of thin wire will be much greater than in the long piece of thick wire, because the mass to be heated is less in the former case. It was found that with two wires of the same material conducting the same current the rise of temperature was inversely proportional to the square of the cross section.

For these reasons it is specified for wiring aboard ship that the copper must have a conductivity not less than 98 per cent. of that of copper chemically pure; the cross-section of the wires is to be such that not more than 1,000 ampères pass per square inch.

Safety fuses depend for their action on the rise in temperature due to resistance of the current. They consist of short pieces of some metal of low fusing point (usually tin, sometimes alloys of bismuth, lead and tin). They are made of such diameter as to be considerably heated by the passage of the ordinary current, and to melt when its strength reaches a certain point above the ordinary.

The dynamos used are mostly compound wound continuous-current machines. The electrical pressure at the terminal points is to be constant, and as the switching on or off of lamps adds or diminishes resistance with a corresponding tendency to slow down or accelerate the machine, the driving engine is so regulated that the power to be developed may fall from full

power to $\frac{1}{10}$ full power accompanied by a variation in the number of turns of only 5 per cent.

Of the two systems of wiring, with single or double leads, each has its advantages. The single-wire system has the advantage of cheapness, and, it is claimed offers less resistance, owing to the ship being used for the return circuit. The double-wire system affords greater security; with a single wire a single fault in the insulation is sufficient to allow a grounding and consequent short circuit through the ship, while with the double-wire system both must develop insulation faults to allow a similar short circuit.

With either system the insulation is of the highest importance. Experience has taught that permanence is a matter to be more sought after than high initial insulation resistance. Many substances perfectly serviceable ashore cannot be used on board ship, on account of changes of temperature, moisture, action of sea water, &c. The substance least affected by these causes is doubtless vulcanized rubber. This, however, must not be brought in direct contact with the wire, as the sulphur contained corrodes the copper. The plan is therefore followed in practice of covering the wire first with a layer of pure rubber, which is followed by a layer of some protecting material, and finally by the vulcanized rubber. The outer covering consists of linen, which is better put on as a serving than as a wrapping, because the latter is liable to leave openings through which the air may attack the insulator.

Connections with branch circuit wires are made by first removing the insulation and passing the branch wire several times around the main conductor, after which the joint is well soldered. No muriate of zinc or muriatic acid is to be used. The joint is next wrapped with pure rubber, and then with several layers of vulcanized rubber which are connected together by means of a solution of caoutchouc. Finally it is wrapped with strips of rubber canvas. Much depends on the care taken by the fitter. The greatest enemy of a joint is moisture, and for this reason many makers place all the connections in water-tight metallic boxes.

A fire may result from a short circuit, which allows the current to reach the return wire without traversing the lamps. If the direct connection so made is of small resistance, the current will grow very strong with corresponding rise in temperature of the conducting wire. Excessive rise is prevented by the safety fuzes when they are well adjusted to the normal strength of the current.

Other faults that may give rise to fires are local reductions in cross-section of the wire or a complete break. The former, which may be the result of corrosion of the wire, offers inreased local resistance, resulting in a corresponding rise in temperature. A break in a wire, which may have either existed in the wire as originally fitted or be the result of external injury, leads to the formation of an arc. Neither of these dangers are under control of the safety fuzes.

A direct connection between the leads of great resistance may also be a cause of fire. Although the current passing through it may be too weak to actuate the safety fuzes, yet the gradual accumulation of heat in a confined place may ultimately lead to ignition.

These dangers are to be averted by the "concentrix system" of J. D. F. Andrews. Only one lead is used, a cable, whose core is an insulated copper wire, forming the outward circuit, while an external iron sheathing conducts the return current. The last is not insulated from the ship, making this system closely related to the ordinary single-wire system. It is claimed that the outer casing affords complete protection to the copper core, and that local attenuation or break will lead to no danger, as a short circuit, actuating the safety fuzes, is at once established. With this system the branch circuits are connected in the usual way as regards the copper wire. Connection is made between the outer casings by special couplings which are said to work well.

On ships carrying petroleum some further dangers exist not found on other ships. When turning any circuit on or off a spark is made at the switch. These must, therefore, be placed above decks or in places where inflammable gases cannot collect. An explosion of gases may be caused by the breaking of a

burning incandescent lamp. These should, therefore, be enclosed in a second air-tight glass globe.

The following directions are deemed sufficient to reduce the dangers attending electric installations to a minimum.

PROPOSED DIRECTIONS GOVERNING THE USE OF ELECTRIC LIGHT ON BOARD OF SHIPS.

1. *Location of Dynamos and Electric Motors.*—These are to be placed as far removed from all compasses as is possible. Their distance from the standard compass is never to be less than 30 feet.

2. *Lead of wires.*—On ships fitted with continuous current dynamos and the single-wire system, no single lead is to approach a compass nearer than 15 feet; with very powerful currents this distance must be greater. If it should be necessary to approach nearer than the given limit, all circuits fed from such wires must be supplied with wires for the return current, which must be led as close as possible to the outward circuit wires, where they approach the compass closely.

3. *Correction of Compass.*—The compass deviations are to be determined as usual with the dynamos at rest. The corrections are then to be applied to different courses with the dynamos in operation, both with all circuits closed and also with those in the neighborhood of the compass cut out. Any change of deviation is to be noted, and corrections applied if found necessary. On ships fitted on the double-wire system the error introduced will be much smaller than on those fitted on the single-wire system. A test should, however, be made with all circuits open.

4. *Leads and Wiring.—a.* The copper used in the wires should have a conductivity which is at least 98 % of that of copper chemically pure, and the cross section must be such that not more than 1,000 ampères flow through an area of a square inch.

b. No wire is to be used smaller than No. 16 B.W.G., except for portable leads, in which the wires composing the cable may be thinner.

c. The insulation resistance is not to fall below 600 megohms per statute mile after immersion in sea water for twenty-four hours.

d. If rubber is used as the insulating material, the wire is to be first covered with a layer of pure rubber; then follows a layer of some separating material; then vulcanized rubber, and finally rubber canvas. The whole is then vulcanized, and is thereupon covered for protection with a serving or a wrapping of saturated hemp, the serving being preferred.

e. Wires insulated with some material other than rubber must pass the above specified tests.

f. The wires are to be so led as to be readily accessible. If covered by wood battens, these are to be fastened by screws, not nails, and care taken to protect them from moisture. If not covered by wood battens, the wires are to be supported by screw clamps, not by staples. Wires are never to be left uncovered, however, except where it is impossible to do otherwise. Cable protected by a metallic casing may be left exposed.

g. Wires exposed to moisture, particularly when at the same time subject to heat, are to be covered by a lead casing or be protected by an equivalent device. This applies to exposed locations and to wires in the vicinity of water-closets, in the engine and boiler rooms, etc.

h. All wires passing through cargo spaces or coal bunkers must be covered by a lead casing secured by iron bands and carried in metal tubes.

i. Wires passing through iron beams or bulkheads are led through hard-wood ferrules; where passing through decks, they are led through metal tubes projecting at least six inches above the deck, to which they are securely fastened and made watertight by wood or fibrous packing.

k. On ships having compartments which are used alternately for cargo and for passengers, the lamp connections are to be made removable in such compartments and provided with an insulating cover. Switchboards and safety fuzes are to be mounted outside of these compartments and to be made secure against being tampered with.

5. *Connecting the Wires.—a.* Joints are to be avoided in the main leads as far as is possible. Where unavoidable, the joint, after careful splicing, soldering, and insulation, is to be protected

by a water-tight box. If the wire is protected by a lead casing, this casing is to be made whole over the joint by soldering.

b. Connections between branch wires and between branch and main circuits are to be carefully soldered and perfectly insulated and made water-tight. Joints in the outward and the return circuit wires should never be placed close together.

c. Rosin is the only flux permissible in soldering.

6. *Hull Connections.*—On ships fitted on the single-wire system all connections with the hull are to be made at accessible places. Connection is to be made by means of brass screws of at least ¾ inch diameter, tapped securely into the hull plates. White metal washers are to be inserted between the wire and the steel, or the wires may be soldered to brass plates.

7. *Switches and Safety-fuses.—a.* A main switchboard is to be fitted in the dynamo room to which all main currents on the ship are led, each of which has a separate switch and safety fuze. If auxiliary switchboards are fitted, they are to be mounted in accessible places, and each branch circuit is to have its cut-out and safety fuze.

b. The switches are to be mounted on a base made of slate or some other incombustible material, and to be so constructed as to either allow the full strength of current to pass or to cut it out completely, without permitting of intermediate positions. The contact faces must be large and their resistance not greater than that of the wire conductor.

c. Safety fuzes are to be fitted for the main circuits as close to the dynamo as possible, and for branch circuits as near their cut-outs as possible.

d. All safety fuzes are to be mounted on slate or similar material, and furnished with a cover of incombustible material. The fuze is to be so arranged as to preclude all possibility of causing fire.

e. The fuze wires are to be made of non-oxidizable metal, and to be so proportioned that they melt when the strength of the current reaches 50 % above the normal, *i. e.*, when the quantity passing per square inch reaches 1,500 ampères. The fuze strips

of the main safety fuzes are to be made of a number of thin strips rather than of one of greater thickness.

f. On ships with the double-wire system, all wires are to have safety fuzes.

8. *Insulation, &c.—a.* After completing the installation, the insulation resistance of all parts of the circuit is to be determined before a working trial is made, and again after a six-hour trial. The insulation is to answer the following requirements:

b. The insulation resistance of the complete system is not to be less than:

500,000 ohms for	25 lamps.
250,000 ohms for	50 lamps.
125,000 ohms for	100 lamps.
25,000 ohms for	500 lamps.
12,500 ohms for	1,000 lamps.

c. For a number of lamps intermediate between those given the insulation resistance is to be similarly proportioned.

d. The insulation resistance of the separate circuits is to follow the same rule.

e. With alternating currents the minimum insulation resistance is to be double that given above.

f. No system is complete without a volt meter and an ampère meter.

9. *Ships carrying petroleum.*—All switches and safety fuzes are to be mounted in places where inflammable gases cannot collect. Lamps installed in dangerous places are to be surrounded by a second air-tight glass globe. All wires in such compartments are to have casings of lead or other material not subject to corrosion by the petroleum.

Discussion.

Passed Assistant Engineer G. W. Baird, U. S. Navy.—These revised rules, which are the outcome of experience, certainly are worthy of careful consideration. The cost of the installation of a plant, under such rules, would be much less than under the

methods employed in our own Navy, and I have no doubt would answer every purpose. It was my fortune to instal the first incandescent plant on board a Government vessel. It was early in the period of ship lighting by electricity, and there were no rules then for my guidance. I do not think I would have considered then, or now, a proposition to use a single wire, and use the hull of the ship for a return; it is taking too much risk for the amount of saving of wire realized. On the other hand, the installation may easily be made so expensive as to discourage ship owners from the use of electric lighting.

A great engineer once said: "It is a good mechanic who knows when a job is good enough." This applies quite as well to ship wiring. It is universally admitted that steel is a better metal than cast iron; it brings a higher price; it is stronger; and yet we continue to make parts of our engines of cast iron; it is *as good for the purpose* as steel. And so it is with wire. The rubber-covered wire specified in the above rules is quite as good as the rubber and lead-covered wires used on board our ships. Where a wire has to pass through a very hot place, or where it is likely to be struck by a jet of steam, the lead covering is advisable. In dry places, in wooden mouldings, a low grade of insulation is sufficient, but with the traditional deluging of our ships of war, it is difficult to designate a dry place on board.

In the matter of making splices and taps, there is difference of opinion between authorities. The objection to the use of muriate of zinc as a flux is, that should there be an excess of acid there will result corrosion, electrolysis, and the wire will be severed at that point, in time. If rosin be employed instead, it may, and frequently does, separate the surfaces which should be joined, which fault is concealed by an envelope of solder; the result is poor contact, and heating at that point. The so-called "water-tight" junction box, an expensive affair, is often used in lieu of splices. The commercial junction boxes have not sufficient joint surface nor enough screws to make a fair joint; they could be made better, but their bulk, weight, and cost would require to be greatly augmented. My own method in splicing is

to strip off the insulation, tin the wire, twist together, use muriate of zinc as a flux, and solder; then apply a weak solution of sal soda, or aqua ammonia, with a sponge, in order to neutralize any excess of acid, and then to wash the joint freely with clean water; then tape with okonite or with Grimshaw splicing compound, and finally with Edison black cotton tape. I have never known one of these joints to give out or to show a defect in any way. They may be immersed indefinitely in water. The expense of such a joint is insignificant when compared with a junction box, and, I believe, is better in every way.

It is inconvenient to lead heavy wires through a ship, on account of their rigidity; it is, therefore, necessary to use either stranded conductors, or a number of conductors of like polarity in the mains. It therefore does not add much to the expense of the job to "crib" the system, *i. e.*, run a conductor or set of conductors from one brush of the dynamo up one side of the ship and back on the other side to the same brush; then a similar circuit from the other brush parallel with the first circuit. Between these (positive and negative) wires the lamps are placed. Now, should one of these wires be corroded or broken on one side of the vessel, there should be no sparking, nor would the lamps go out, for being fed from both directions they will continue to glow. It will be observed, however, that lamps so affected will glow below candle-power, and a careful engineer will quickly discover and repair the damage. It is my practice to follow this "cribbing" wherever possible, and my opinion is that it should be a rule.

There is but scant reference in any rules as to the proper place to pierce a ship's beam or a joist in a house. I have seen the top edges of floor joists notched by two-inch openings, right in the middle of the joist, for the passage of a molding for an electric wire. It was "concealed" work. The strength of those joists was diminished about 30 per cent. A very much better way (and which I advised on that occasion) was to pierce the joists on their neutral axis and string the wires through the tubes, abandoning the molding. The floor above and the plaster below protected the wires from accidental blows or knocks at all times.

In ships we often see the moldings close up under the decks, and the beams either pierced close to the upper flange, or else the deck plank bored for the passage of the wire. The latter is dangerous, as a nail, spike or bolt driven from above would quickly ground it. The frequent and rapid transportation of weights along a ship's deck makes it imperative that the conductors be protected by some covering, but the covering should not be exaggerated. The *weight* (an important item) of the lead covering and of the great wooden molding might profitably give way to something lighter and less bulky. The vulca wire ducts, which are about as strong as our speaking tubes, present themselves as formidable rivals. These ducts may be threaded like pipe, screwed into the beams, and form continuous conduits, strong and safe. So long as parallel positive and negative wires in close proximity pass near the compasses there does not appear to be any appreciable disturbance; but if the dynamos are near to the compasses, the latter will be affected by the field and will be greatly disturbed. A great mistake has been made on board some of our Navy ships in the location of their dynamos. If we did not have accomplished navigators I fear some of our best vessels would have been lost.

The best place—for many reasons—for the dynamos is in the engine room. It is always far removed from the compasses, and as the management of a dynamo is simple, an intelligent coal-heaver can learn to run it in a few hours. The practical application of electricity is not an occult science. The most ordinary intellect can readily grasp it and utilize it. Sir William Thompson has said that a man to run an electric plant needs be nine-tenths engineer and one-tenth electrician. In my own experience I find that the engine which runs the dynamo gives nearly all the trouble, the dynamo but little. The noise of the dynamo is drowned by the greater sounds from the engine. The heat and the steam should be confined as nearly as possible to the engine room. When the dynamo is in the engine room it requires no additional attendants; when on deck it requires a corps of attendants in addition to the engine-room force.

PASSED ASSISTANT ENGINEER W. N. LITTLE, U. S. NAVY.—In the proposed directions governing the use of electric light on board of ships, the location of the dynamos and electric motors receives first attention. They should be as far removed from all compasses as possible.

So far as relates to the compasses, no objection can be found to placing the dynamo in the engine room, or in a compartment adjacent thereto. On the contrary, a positive advantage would be gained by such a change from the usual position on the berth deck. The proximity of a mass of iron in the motive machinery of the ship would operate to the advantage of the compasses. There are other reasons for the choice of the engine room location, and these reasons seem to have weight in the mercantile marine, for I have seen dynamos installed in the engine rooms of the O. and O. steamer *Coptic* and the Plant Line steamer *Olivette*. In unarmored ships of war it is attaching but slight importance to the electric search-light plant to place its dynamos above the water line and protective deck, where it will be exposed to the fire from machine guns, inviting destruction to the ship's torpedo defence.

Space in the engine room or in a near compartment could be easily provided. The foundations of the dynamo engine could be made more solid and the annoyance due to vibrations less ened: with vertical engines this point is an important one.

Now in respect to attendance on the electric-light machinery:

A dynamo has but few working parts to become deranged compared with its driver, the dynamo engine. It is the latter which requires the chief attention and care, and should have it from skilled hands under professional control. A large amount of piping and its repair, required by the present position of the dynamo could be avoided; also the services of several men; for the oilers of the watch could lubricate, and the machinist could attend to the dynamo and its engine in addition to his other duties; or if a regular dynamo attendant is deemed necessary, he should also be available for other service in the line of his skill with tools. He should belong to the Engineer's De-

partment, and the entire electric-lighting plant should be under the charge of the Chief Engineer.

CHARLES H. DAVIS, C. E., ELECTRICAL AND MECHANICAL ENGINEER.—In the article "Electric Lighting of Ships" there are several points of interest to those desiring improvement in all lighting and power systems where electricity is used.

By the use of some form of iron-clad dynamo the effect of magnetism on the compass is reduced; this is, however, not necessary when the dynamo is far enough away from the compasses.

To avoid induction effects from the current in the wires on the compass, the two poles should be as close together as consistent with good insulation and balanced as to relative distance from the compass.

The error of the compass should be determined both with the dynamo fully loaded, and when there is no current except that in the fields which is necessary to maintain the normal potential of the machine.

The best practice in the United States is against the use of the single-wire system, making the ship the return. Defective lighting is almost sure to result from its use.

For practical purposes from an electric standpoint the "concentrix system" amounts to the same as the single-wire system and has the same objections. From a mechanical point of view it is somewhat better.

The use of rubber-covered wire will not in itself be a safeguard, and it is my opinion that a cotton insulation with the use of plenty of white lead will give better results.

Lead-covered wire has objections which I think more than counterbalance all points in its favor except under peculiar circumstances where there is nothing else to be used.

The avoidance of joints is recommended throughout the plant, even in branch circuits; connections should be made only at fuse blocks, and no fixture fuses should be used. This method, called the "pocket system" or "panel-board system," prevents short circuits, and if the wires from these pockets to each outlet are so arranged that they may be easily removed, the system will be most flexible and can be repaired at a minimum cost.

All fuses and switches should be mounted on slate or porcelain and located where there is no possibility of their causing fire.

Double-pole switches should be used throughout on all ship work.

As to insulation of the system: authorities of equal standing differ so widely that it is impossible to set a standard, and good judgment is more necessary in this than in almost any other part of the system. In general, a system running on 110 volts should have an insulation of not less than 50,000 ohms irrespective of the number of lights, and this should be maintained very carefully.

Wires can be run in molding or in tubes as desired, though the former is the easier kind of construction to do neatly in a vessel. Everything should be as tight as possible to prevent moisture from lowering the insulation.

AUGUSTUS NOLL, ELECTRICAL ENGINEER.—Instead of making connections between the branch circuit wires in the manner stated, it would be better to insert a cut-out and box of approved pattern with the entrance and exit for the wires water-tight. There are less chances of trouble by this method than by making a joint by splicing same. The cut-out in this box will, at the same time, answer as the fixture cut-out.

Regarding the concentrix cable: To be properly constructed, the material or insulation between the copper core and the external covering or pipe must be fibrous, and easily carbonizable; but being fibrous, it will absorb moisture, and, as the pipes are made of metal, there is bound to be condensation, the moisture will be absorbed by the fibrous insulation, and in time will create partial short-circuits; these, in some cases, will not be sufficient to fuse the metal strips on the cut-outs. If these metal tubes are fastened in place without sufficient play allowed for the laboring or tossing of the vessel, the joints in time will become loose, and the liability of trouble from moisture will be increased.

Regarding vessels carrying petroleum, etc.: It would be better to center all the switches in the dynamo room instead of above decks; being compelled to place the main line switches in the dynamo room, there is no reason why all the switches cannot be

placed together, thereby lessening the chances of trouble. A switch-board under these conditions can be made as follows: Mount all the cut-outs and switches on porcelain bases, the porcelain to be highly vitrified; then fasten same to back board made of slate (porcelain is an absolute non-absorbent, while slate is not). All the working metal parts on both cut-outs and switches should be extra heavy, and the switches should have 50 per cent. metal in excess of their rated carrying capacity, and all slide and movable connections thereon should be locked, so that the vibrations of the vessel will not affect them to an extent where the connections may become loosened. This can be done by means of tension springs. The whole switch-board can then be placed in a water-tight and fire-proof box or closet. From this closet can be run a pipe of sufficient diameter to the upper deck, on the end of which can be placed a ventilator, and by means of it the gas which may generate in the switch closet can be exhausted.

Regarding 5 *b* of rules: It is better to insert a cut-out and box similar to that noted above.

Regarding 7, 7 *b* and 7 *c* of rules: All switches and cut-outs should be double poled and mounted on highly vitrified porcelain instead of slate. Slate absorbs moisture, and also contains a metallic substance, which in numerous cases has caused trouble. All switches and cut-outs centered and mounted on the switch-board should first be mounted on porcelain, and then fastened to the slate or marble back-board with sufficient air space between the back of board and surface of the side of the vessel. All switches and cut-outs should be nickel or silver plated, and all working movable parts should be tinned to prevent oxidation.

All fuse wires should have copper terminals, and the terminals should be tinned, thereby securing clean connections.

Regarding 4 *f*, *g*, *h* and *i*: The Vulca Electrical Wire Duct System will give better satisfaction than the materials mentioned.

Note: The single-wire system offers no advantage in comparison with the return-wire system, not even in cheapness, as the cost of maintenance and other inconveniences more than offset the difference in first cost. The liability of the danger of fire, etc., is greater by far in the first mentioned system.

XXXII.

RADIAL VALVE GEARS.

The following is a translation of a series of articles on the above subject which appeared in the "Zeitschrift des Vereines Deutscher Ingenieure" in October and November of 1889. The articles referred to were an abstract of a prize essay by Curt Fränzel, Marine Engineer, of Kiel.

Most writers on this subject have either attempted to develop accurate equations for the curves representing the valve travel, which are of little or no practical use when determined, or they have confined themselves to descriptions and recommended design by trial. A notable exception was the paper contributed to the JOURNAL on the Marshall valve gear, by Passed Assistant Engineer I. N. Hollis, U. S. Navy.

In the following the writer contents himself with giving the equations to the approximate Zeuner valve circles, and then eliminating the errors due to the angularity of the swinging parts of the mechanisms.

Before entering on a discussion of the different varieties of radial gear it is, therefore, necessary to find simple algebraic expressions for these errors.

Error due to the angularity of a rod swung from mid-position: Rod AB, Fig. 1, Plate 1, receives a lateral movement, m; then n is the error due to its angularity. It is given by the portion of the perpendicular erected at C intercepted between BC and the arc of a circle struck with A as center and AB as radius. This construction is employed to determine the valve travel from the curve described by the free end of the valve connecting rod; *e. g.*, let the free end of the rod AB (Fig. 2, Plate 1) describe the curve indicated. Required the distance the valve has moved from mid-position for point P. About A, with radius AB, describe an arc of a circle; drop a perpendicular upon the axis of X from P; then PP_1 is the distance required.

Analytically we proceed as follows:

From Fig. 3. $n = \rho - \sqrt{\rho^2 - m^2}$,

and $\rho = \frac{m^2}{2n} + \frac{n}{2}$.

For comparatively small vibrations (m), like those occurring in the mechanisms to be dealt with, $\frac{n}{2}$ is small in comparison with $\frac{m^2}{2n}$ and ρ, and may be neglected, wherefore we have,

$$\rho = \frac{m^2}{2n},$$

$$\text{and } n = \frac{m^{2*}}{2\rho} \quad \ldots\ldots\ldots \quad (1)$$

A problem in connection with the foregoing is the following: Required the radius of a circle, which, with a given m, will have a versed sine n greater or less by a given quantity than that of another arc of known radius. In Fig. 4, Plate I, given a circle A with certain values of m and n: required a circle with swing m and versin $n \pm n_1$.

According to (1) . . . $n = \frac{m^2}{2\rho}$.

n_2 the versin required $= n \pm n_1$; therefore

$$n_2 = \frac{m^2}{2\rho} \pm n_1 = \frac{m^2}{2\rho_1}.$$

Where ρ_1 is the radius of the circle required.

$$\text{Then } \rho_1 = \frac{m^2\rho}{m^2 \pm 2\rho n_1}. \quad \ldots\ldots\ldots \quad (2)$$

Application of the above formulæ:—Required the error due to the angularity of a connecting rod. (Fig. 5, Plate I.)

$$m = R \sin \omega;$$

$$\text{hence } n = \frac{R^2 \sin^2 \omega}{2l}; \text{ making } l = qR,$$

$$\text{we get } n = \frac{R \sin^2 \omega}{2q}.$$

* *I. e.*, a parabolic in place of a circular arc.

For example, for the engine to which the valve gears to be considered are to be applied in succession, where $q = 5$,

$$n = \frac{R}{10} \text{ for } \omega = 90°.$$

Another problem that occurs frequently is the following: Point A of the rod AB (Fig. 5, Plate 1,) moves in the axis of X, *i. e.*, is neither raised nor lowered. Point C is the point of application of a swinging link of a length to be determined. B is the point of application of rod r, D being fixed. How long must r_1 be, so that D may remain stationary (or approximately so)?

Let point B be moved to B_1. Let the swing and the corresponding versin then be m and n. Link r_1 vibrates through the same distance, m (neglecting the angularity of AB); its n_1 is given by

$$n_1 = \frac{AC}{AB} \times n.$$

$$\text{From (1) } r_1 = \frac{m^2}{2\frac{AC}{AB}n}.$$

$$\text{But } r = \frac{m^2}{2n}; \text{ therefore } r : r_1 = AC : AB. \quad . \quad . \quad (3)$$

This may be shown in another way: Point C travels approximately on an ellipse with the semi-axes r and $r\frac{AC}{AB}$. The radius of curvature for point C in mid-position is therefore

$$r_1 = \frac{r^2}{r\frac{AC}{AB}} = \frac{r}{\frac{AC}{AB}} \text{ or, as before,}$$

$$r : r_1 = AC : AB.$$

It may become necessary to substitute for straight guides or for swinging links of great length parallel motions or mechanisms for producing motion on arcs of circle.

The form of parallel motion shown in Fig. 6, Plate 1, is to be transformed into one producing circular motion.

ρ and ρ_1 are radius rods of equal length. AC is the connect-

ing link. The guided point B is at the middle of the length AC. It is required to guide B on the arc of a circle with radius ρ_2.

Let the mechanism be swung to the right a distance m; then the n's become $= \frac{m^2}{2\rho}$, being equal but in opposite directions. A circle of radius ρ_2 would have an $n_2 = \frac{m^2}{2\rho_2}$. This should be the versin of B. As point B is in the middle of AC, point C should be raised a distance $\frac{m^2}{\rho_2}$ in addition to the distance $\frac{m^2}{2\rho}$. (Assuming that the center of the circle on which B is to travel lies above AC.) According to (2) we therefore obtain for the new length of ρ,

$$\rho_1 = \frac{m^2\rho}{m^2 + 2\rho\frac{m^2}{\rho_2}} = \frac{\rho\rho_2}{\rho_2 + 2\rho}.$$

For example:

$$\rho = 10 \text{ cm.}, \qquad \rho_2 = 50 \text{ cm.};$$

$$\text{therefore } \rho_1 = \frac{10 \times 50}{50 + 20} = 7 \text{ cm.}$$

This combination guides accurately only for a short distance, as the upper radius rod becomes too small and the errors soon become too great. The same end may be gained, leaving the radius bars unchanged, by shifting B on AC.

From Fig. 7, Plate I,

$$n_1 = \frac{m^2}{2\rho_1}.$$

The arc to be described is to have an

$$n_2 = \frac{m^2}{2\rho_2}.$$

$$n_2 : n_1 = s : l,$$

$$\text{and } s = \frac{n_2 l}{n_1}.$$

Substituting the values of n_2 and n_1

$$s = \frac{\frac{m^2}{2\rho_2}\,l}{\frac{m_2}{2\rho_1}} = \frac{l\rho_1}{\rho_2}.$$

Example: $l = 2.5$ cm., *i. e.*, $AC = 5$ cm.; ρ and $\rho_1 = 10$ cm.; $\rho_2 = 50$ cm. Then

$$s = \frac{2.5 \times 10}{50} = 0.5 \text{ cm.}$$

Figure 8, Plate I, is the mechanism used by Kirk in his valve gear. AC is broken as shown and β is 30° in the example given. This motion guides well in a straight line bisecting β, *i. e.*, with B in the middle of the length AC.

We will now deduce a formula for the motion employed by Brown in his gear. Point A, Fig. 8, Plate I, is to be guided in a straight line.

$$OA = l; \quad OB = l_1.$$

The error due to the angularity of the length OA with a vibration of amplitude m is

$$n = \frac{m^2}{2l}.$$

This is to be corrected by the rod BC, jointed at B. Length OB on l has

$$n_1 = \frac{m^2}{2l^2} \times l_1 = \frac{m^2}{2l} \times \frac{l_1}{l}.$$

The difference between n_1 and n is

$$n - n_1 = n_2 = \frac{m^2}{2l}\left(1 - \frac{l_1}{l}\right).$$

This difference must be neutralized by $BC = \rho$ swinging through $m_1 = m\frac{l_1}{l}$. From (1) we get

$$\rho = \frac{m^2\left(\frac{l_1}{l}\right)^2}{2\left[\frac{m^2}{2l}\left(1 - \frac{l_1}{l}\right)\right]} = \frac{l_1^2}{l - l_1}.$$

Let $AB = \frac{1}{3}l$ or $l_1 = \frac{2}{3}l$, then

$$\rho = \frac{\left(\frac{2}{3}l\right)^2}{l - \frac{2}{3}l} = \frac{4}{3}l.$$

This combination, also, may be made to produce motion on a circular arc. For $\rho = \frac{l_1^2}{l - l_1}$, A moved in a straight line; it is now to describe an arc of curvature ρ_1. Let n_3 for this circle $= \frac{m^2}{2\rho_1}$.

This is to be added to n_2:

$$n_2 + n_3 = \frac{m^2}{2l_1}\left(1 - \frac{l_1}{l}\right) + \frac{m^2}{2\rho_1}$$

$$= \frac{m^2}{2}\left(\frac{1}{\rho_1} + \frac{1}{l_1} - \frac{1}{l}\right).$$

$$\text{Whence } \rho = \frac{m^2 l_1^2}{l^2 m^2\left(\frac{1}{\rho_1} + \frac{1}{l_1} - \frac{1}{l}\right)}$$

$$= \frac{l_1^2}{l^2\left(\frac{1}{\rho_1} + \frac{1}{l_1} - \frac{1}{l}\right)}.$$

Substituting as above $l_1 = \frac{2}{3}l$

$$\rho = \frac{4}{9\left(\frac{1}{\rho_1} + \frac{1}{3l}\right)} = \frac{4 \times \rho_1 \times l}{9l + 3\rho_1}.$$

Radial valve gears may be divided, according to the part from which motion is derived, into—

1. Gears actuated by one eccentric.
2. Gears actuated by the connecting rod.
3. Gears actuated by the connecting rod and piston rod.

The first class comprises, among others, the gears of Fink, Hackworth, Angström, Marshall, Klug and Brown. This class might also be made to include the valve gear of Heusinger v. Waldegg, which derives its motion from an eccentric and the piston rod.

Class 2 comprises the gears of Brown No. 2 and Brown No. 3 (known as Joy's gear).

Class 3 comprises the gears of Joy and Kirk.

To be able to make a comparison between these different gears, they were all—with a single exception—applied to the same en-

gine, an inverted vertical, two-cylinder compound, with piston valves at the sides of the cylinders. The dimensions bearing on the design of the valve gear are:

Throw of crank, . . .	$R =$ 20 cm.
Length of connecting rod, .	$5R =$ 100 cm.
Distance from center line of cylinder to center of valve, . .	$Q =$ 44 cm.
Valve stem crosshead above center line of shaft, . . .	$=$ 112 cm.
Steam lap, ...	$e =$ 27 mm.
Exhaust lap, . . .	$i =$ 10 mm.
Lead, ...	$v =$ 3 mm.

I. VALVE GEARS ACTUATED BY ONE ECCENTRIC.

The method of determining the approximate Zeuner diagram and then making corrections is applied first to the valve gear (Figs. 15 and 16, Plate 2) of Pius Fink. For marine engines, whose valve motion is to be as exact as possible, only that form of this gear is applicable in which the pivot, the motion about which gives port opening to the valve, is on the link.

The motion of any point C_1 of the link may be considered as composed of two movements. If C_1 was coincident with N, we should have simply the travel due to the throw of the eccentric. Its Zeuner diagram would be represented by Fig. 11, Plate 1. The co-ordinates of the center of the Zeuner circle would be $A = \frac{r}{2}, B = o$. The angular advance in this case would be 90°. The error due to the angularity of the eccentric rod will, however, result in making the upper valve circle smaller than the lower.

If now C_1 is moved a distance NC_1 from the pivotal point, it will receive, in addition to the motion imparted before, the link being perpendicular to the eccentric rod, a second motion in the same direction, which is due to the oscillation of the eccentric rod.

The extent of the first motion is given by the expression $r \cos \omega$ (r is the throw of the eccentric, ω the angle the crank, or

the eccentric, has moved through); the extent of the second motion is given by $r \sin \omega \frac{u}{l}$ ($u = NC_1$ and $l = EN$). The valve circle corresponding to the latter motion is shown in Fig. 12, Plate I; it is at an angle of 90° with the former. While the diagram giving the first motion is unalterable for ahead or backward motion of the engine and for all degrees of expansion, the latter changes with u, and it is revolved through 180° when u is measured on the other side of N. As C_1 partakes of both movements simultaneously, their algebraic sum—the direction of the movements being identical—will give the final motion of C_1. The travel ξ then will be

$$\xi = r \cos \omega + r \sin \omega \frac{u}{l},$$

the equation to Zeuner valve circles, the co-ordinates of the centers being

$$A = \frac{r}{2}; \quad B = \frac{r\omega}{2l}.$$

If the ratio of length of eccentric rod to eccentric radius is made the same as that of the connecting rod of the engine to the crank, the errors due to angularity will have a fixed ratio. The valve will therefore travel similarly to the engine piston, and the steam supply to both ends be regulated in accordance with the piston travel. The cut-off and the direction of motion of the engine are therefore dependent on the magnitude and direction of the second movement, *i. e.*, on the position of C_1. The further C_1 is away from N, the later will be the cut-off; and the engine will go ahead or back according as C_1 is on one side of N or the other.

The valve circle giving the first of the movements composing the resultant motion of C_1 will be known as the *lead circle;* the second as the *cut-off circle;* the resultant valve circle will be called the ξ *circle.*

These diagrams are applicable to all the different forms of gears. We will therefore dwell upon them a little more at length, as it is of importance in the design of these gears to ascertain the size of the cut-off circle.

In the present case we have given particulars of the valve: steam lap = 25 mm. = e; lead = 3 mm. = v. Required the lead and cut-off circles. The lap and lead determine the lead circle at once:

$$A = \frac{e + v}{2}, \quad B = o.$$

The locus of the centers of all ξ circles is therefore a straight line parallel with the axis of X at a distance from it $= \frac{e + v}{2}$. Cut-off is to take place at P (Fig 13, Plate 1). Graphically the solution is a very simple one. Draw OP, bisect it, and erect a perpendicular at P_1; its intersection with the locus MN is the center of the resultant valve circle, and $MN = OM_1$ is the radius of the cut-off circle required. This radius will be designated by the letters f_1, f_2, &c.

Analytically the solution proceeds as follows:

The equation of the line $P_2 P_3$ as tangent to the circle with radius $r = \frac{e}{2}$ is

$$yy_1 + xx_1 = r^2$$

where y_1 and x_1 are the co-ordinates of P_1.

$$\text{Hence } x = \frac{r^2 - yy_1}{x_1}.$$

$$\text{Moreover } x_1 = \sqrt{r^2 - y_1^2}.$$

Substituting this in the above formula, and determining the value of x for $y = \frac{e + v}{2} = A$, we shall have

$$x = \frac{r^2 - \frac{e + v}{2} y_1}{\sqrt{r^2 - y_1^2}}.$$

Generally, substituting for y_1, $y_1 = qr$, we finally have, since $r = \frac{e}{2}$

$$x = B = f = \frac{e - (e + v)q}{2\sqrt{1 - q^2}} = [e - (e + v)q] \frac{1}{2\sqrt{1 - q^2}}.$$

The following table contains the values of q and of $\frac{1}{2\sqrt{1-q^2}}$ for points of cut-off ranging from .2 to .8:

Cut off.	q.	$\frac{1}{2\sqrt{1-q^2}}$.
.2	+ .6	.6250
.25	+ .5	.5773
.3	+ .4	.5456
.35	+ .3	.5241
.4	+ .2	.5109
.45	+ .1	.5025
.5	± 0	.5000
.55	− .1	.5025
.6	− .2	.5109
.65	− .3	.5241
.7	− .4	.5456
.75	− .5	.5773
.8	− .6	.6250

From this table we shall get, for $e = 27$ mm., and $v = 3$ mm.,

Cut-off

.5 . . . $f_1 = 1.35$ . . . $2f_1 = 2.7$
.4 . . . $f_2 = 1.05$ . . . $2f_2 = 2.1$
.3 . . . $f_3 = .80$ . . . $2f_3 = 1.6$

These values of f hold for all other types of valve gear.

For the Pius Fink gear under consideration, we have:

r of the eccentric $= e + v = 30$ mm.

u_1, *i. e.*, the distance of C_1 from N for .5 cut-off,

$$u_1 = 2f_1\frac{l}{r};\ u_2 = 2f_2\frac{l}{r};\ u_3 = 2f_3\frac{l}{r}.$$

For backing, cutting off at half stroke,

$$-u_1 = -2f\frac{l}{r}.$$

As in the present case, $\frac{l}{r} = 5$,

$u_1 = 2.7 \times 5 = 135$ mm.
$u_2 = 2.1 \times 5 = 105$ "
$u_3 = 1.6 \times 5 = 80$ "
$-u_1 = -2.7 \times 5 = -135$ "

The remaining lengths may be taken from the drawing. Fig 16, Plate 2, shows the complete gear as designed to cover the present case. The link is guided vertically by means of a slot guide in order to avoid the irregularities due to the swinging link 1, Fig. 15. The reversing arm u_2 is very short, which introduces errors into the diagram. The latter, Fig. 17, gives valve travels proportional to the piston travel only in the lower valve circle.

In order to reach the valve stem, which is at the side of the cylinder, a rockshaft is introduced; the eccentric is therefore to be set with the crank.

HACKWORTH GEAR.

(Figs. 18 to 26, Plate 2.)

To form the approximate diagram:

Lead circle: This is determined by r and by the ratio of the lengths $G_1G_2 = l_2$ and $O_1G_2 = l$; the polar co-ordinates of this circle are therefore

$$\xi_1 = r\frac{l_2}{l}\cos\omega \text{ and } A = \frac{r}{2}\times\frac{l_2}{l}.$$

Cut-off circle: If the slotted link guiding G_2 was placed horizontally, the motion of G_2 in the axis of the guide would be that represented by a valve whose polar equation is $\xi_2 = r\sin\omega$. This motion of G_2 would not, however, result in any motion of the valve. If the guide is now turned through an angle a, these horizontal movements will produce motion vertically equal to $r\sin\omega\tan a$. This is transferred to point G_1 in the ratio of $O_1G_1 : O_1G_2 = \frac{l_1}{l}$. The polar co-ordinates of the cut-off circle are therefore given by the equation

$$\xi_2 = r\sin\omega\tan a\frac{l_1}{l}$$

$$\text{and } B = f = \frac{r}{2}\tan a\frac{l_1}{l}.$$

The two equations combined give for the equation of the *resultant valve circle.*

$$\xi = \frac{r}{l}(l_2\cos w + l_1\tan a\sin w).$$

Determination of the errors.—The error due to the angularity of l is $n = \frac{r^2}{2l}$, by equation (1).

This introduces no irregularity into the lead circle, but it does into the cut-off circle; for in consequence of it the travel of G_2, a and a_1 (Fig. 18, Plate 2) to the left and right of mid position become unequal, being less toward the left by the amount of n and greater toward the right by the same amount. This produces an error vertically equal to $n_1 = \frac{r^2}{2l} \tan a$, and this becomes for G_1, $n_2 = \frac{r^2}{2l} \tan a \frac{l_1}{l}$.

To be added to this is the error due to the valve connecting rod; this is $n_3 = \frac{r^2}{2S}$, where S is the length of this rod. These errors are in the same direction for right-handed rotation, in the opposite direction for left-handed rotation, and are therefore in this case greatest for ahead motion of the engine. The upper valve circle will therefore be greater than it should be, while the lower one will be too small (Fig. 20, Plate 2). This gear should therefore be used only with left-handed rotation of the engine, or, if the rotation is right-handed, steam should be taken on the inside of the valve. (See Fig. 21, Plate 2.) Rod l introduces a second error due to the swinging of point G_2 about O_1 as it travels in the guide. To take account of this would make the calculation too complicated, as it is different for every different value of ω and a. It may be neglected if a is kept small.

The ratio $\frac{l_1}{l_2} = 2$ or $\frac{l_1}{l} = \frac{2}{3}$ is one that will suit the present case. This gives $r = 3(e + v)$;

and $\tan a = \frac{2fl}{rl_1} = \frac{3f}{r} = \frac{f}{e+v}$, *i. e.*, a is less than 45° for all grades of cut-off.

For the design of the gear in the present case we have

$$r = 3(e + v) = 9 \text{ cm.}$$

$$\tan a_1 = \frac{f_1}{e+v} = \frac{1.35}{3} = 0.45.$$

$$\tan \alpha_2 = \frac{f_2}{e+v} = \frac{1.05}{3} = 0.35.$$

$$\tan \alpha_3 = \frac{f_3}{e+v} = \frac{0.8}{3} = 0.266.$$

Q = distance of center of cylinder to center of valve chest
= 44 cm., and

$$l = \sqrt{\left(44 \times \frac{3}{2}\right)^2 + 9^2} = 66.8 \text{ cm.}$$

$$l_1 = \frac{2}{3} l = 44.53 \text{ cm.}$$

These two lengths are best found by construction. Fig. 19, Plate 2, is laid down with these dimensions.

The arrangement of Hackworth's gear just discussed has the disadvantage of necessitating a large eccentric and of occupying much space in the width of the engine. For this reason the arrangement shown in Figs. 22 to 26 is frequently adopted. In this case the guide is placed between the eccentric and the working point, and ordinarily l_1 is made equal to $l_2 = \frac{l}{2}$.

Lead circle: $\xi_1 = r \cos \omega$; $A = \frac{r}{2}$.

Cut-off circle: $\xi_2 = 2\, r \tan a \sin \omega$; $B = r \tan a$.

ξ *circle:* $\xi = r \cos \omega + 2\, r \tan a \sin \omega$.

Construction: $r = e + v = 3$ cm.

$$\tan a = \frac{f}{e+v}, \text{ as for the other form of gear.}$$

This form of Hackworth gear has the advantage over the one previously considered in being more compact, but the disadvantage, that any wear of the guide produces errors which are doubled at the point of attachment of the valve-connecting rod.

The *errors* are: n of rod $l_1 = \frac{r^2}{2l_1}$,

$$n_2 \text{ of the valve-connecting rod} = \frac{r^2}{2S}.$$

These errors are smaller than those for the first arrangement, as r is only one-third of the r of the first form. The error due

to the angularity of l_1, which takes effect in the direction of motion of the valve, is now, however, doubled and equal to

$$n_1 = \frac{r^2}{l_1} \tan a.$$

Both forms are consequently of equal merit as regards steam distribution and give a valve motion corresponding with the motion of the piston itself only when taking steam on the inside.

All radial valve gears to be discussed, with the single exception of the Kirk gear, may be considered as modified Hackworth gears. The only difference lies in the means adopted for correcting the errors inherent in the Hackworth form.

ANGSTRÖM GEAR.

This differs from Hackworth's only in the substitution of a parallel motion for the straight guide. The parallel motion is of the form shown in Fig. 14, Plate I.

MARSHALL GEAR.

Figures 26 to to 31, Plate 3.

This gear differs from the first form of Hackworth's only in that the straight guide is replaced by a swinging link. The author confines the name "Marshall Gear" to that form having the point of attachment of the valve rod between the eccentric and the radius rod. The valve circle is obtained precisely as for the Hackworth gear.

Errors and their correction.—The error due to the angularity of the valve rod is compensated for, according to (3), by making ρ, the radius rod, satisfy the proportion $S : \rho = 3 : 2$; *i. e.*, if l is divided as it was in the Hackworth gear. Hence $\rho = \frac{2}{3}S$; and as $S = 112$ cm.,

$$\rho = 74 \text{ cm.}$$

The diagram given by the gear laid down from these dimensions is better than that of the Hackworth gear (see Fig. 27): the errors are now negative, the upper valve circle being smaller than it should be, the lower too great.

The prinicipal error is that due to the angularity of l.

$$n_2 = \frac{r^2}{2l} \tan \alpha.$$

It is proportional to tan α, and we must therefore decide for what value of α we wish to correct.

The correction is here made for α_1, which is that value of α giving cut-off at half stroke.

$$\tan \alpha_1 = \frac{f_1}{e+v}; \; f_1 = \frac{e}{2}; \; r = 3(e+v); \; \frac{l_1}{l} = \frac{2}{3};$$

$$\text{therefore } n_2 = \frac{[3(e+v)]^2}{2l} \times \frac{e}{2(e+v)} = \frac{9}{4}\frac{(e+v)e}{l}.$$

$$\therefore n_2 = .27 \text{ cm.}$$

Point 7 at which G_2 has arrived must therefore be lowered this amount, which is done, according to (2), by making the length of the radius rod

$$\rho_1 = \frac{m\rho^2}{m^2 \pm 2\rho n_1}.$$

The minus sign is to be used in the present case. Furthermore, $m = r = 3(e+v)$; $\rho = 74$ cm., and $n_1 = 0.27$ cm.

Hence $\rho_1 = 140$ cm.

As a radius arm of this length is inapplicable, because it would collide with the cylinders on being put in position for reversing, its length was taken equal to that of the valve rod $= S$. The resulting diagram, Fig. 28, though not giving a valve travel proportional to the piston travel, is a close approximation to the Zeuner diagram, and goes to show that the errors inherent in the Hackworth gear may be eliminated, at any rate for motion ahead; for backing they appear intensified.

In order to equalize the cut-off two practical methods offer themselves. We may either change the laps of the valve, or we may decrease the travel of G_2 below the horizontal axis, *i. e.*, decrease the upper valve circle—a corresponding increase taking place on the other side—by making the radius arm shorter. The first method is inapplicable, as we wish to make use of the various gears without altering the original dimensions of the valve. With valve travel proportional to the actual piston travel, ξ is less for

the point of the diagram corresponding with the extreme swing of ρ to the left (point 7), than when the travel is proportional to the piston travel with infinite length of connecting rod, by .3 cm. This is the difference for G_1; for G_2 this becomes $.3 \times \frac{3}{2} = .45$ cm. This, therefore, is the amount by which the n previously calculated should be increased for $m = 3\,(e + v) = r$.

$$\rho_2 = \frac{m^2 \rho_1}{m^2 \pm 2\rho_1 n} \text{ (plus sign to be used.)}$$

$$m = r\,;\ \rho = 140 \text{ cm.}\,;\ n = .45 \text{ cm.}\ \therefore\ \rho = 54 \text{ cm.}$$

However, for the point we are considering a is less than that used in the previous calculation. It would be well, therefore, to compensate for this by making ρ still smaller. The diagram, Fig. 29, constructed with $\rho = 50$ cm., gives a good result.

KLUG GEAR.

Figs. 31 to 37, Plate 4.

This differs from the preceding only in that the point of attachment of the valve rod is beyond the radius rod, corresponding with the second form of the Hackworth gear. The diagram is, therefore, obtained in the same way. Certain measurements governing the construction are given by Mr. Klug. They furnish satisfactory results, but are not applicable to all cases. According to these measurements,

$$l_1 = 6.87\ r\,;\ l_2 = 3.435\ r\,;\ \rho = 9\ r.$$

These proportions are unsuitable in our case; we cannot, for example, make $r = (e+v) = 3$ cm., for $l = 44$.

We shall, therefore, assume the ratios adopted for the second Hackworth form.

Errors of the mechanism and their correction.—The error due to the valve rod is corrected according to (3) by making

$$\rho = 2S, \text{ as } \frac{l_1}{l} = \frac{1}{2}.$$

The error measured vertically due to the angularity of the eccentric rod l is $n_1 = \frac{r^2}{2l_1} \tan a$. For cut-off at .5,

$$\tan \alpha = \frac{f_1}{e+v} \quad \therefore n_1 = \frac{(e+v)f_1}{2l_1}.$$

But $e + v = 3$ cm., $f_1 = 1.35 = \frac{e}{2}$ and $l_1 = 22.5$; consequently

$$n_1 = .09 \text{ cm.}$$

The length of the radius rod must be such as to counteract *the sum* of these errors. (In the Marshall gear we had to correct for the difference as the inclination of the radius arm was reversed.)

$$n_1 = \frac{(e+v)f_1}{2l_1} = \frac{(e+v)e}{2l}; \text{ for}$$

$$l_1 = \frac{1}{2}\,l \text{ and } f_1 = \frac{e}{2}.$$

Furthermore $n_2 = \frac{(e+v)^2}{4S}$ (for we found above that to equalize the error due to valve rod S, we had to make the radius rod $= 2S$, for $\frac{l_1}{l} = \frac{1}{2}$). We therefore have for the final length of the radius rod $\rho = \frac{(e+v)2 \times S \times l}{(e+v)l + 2eS}$. Substituting numerical values, $\rho = 40$ cm. The resulting diagram is seen in Figs. 32 and 33, Plate 4.

Again, to make the valve travel proportional to the actual piston travel at the two ends, G_1 must be raised by an amount equal to $\frac{1}{2} \times .3$ or .15 cm. We must be careful in properly making this correction, as any error made will be doubled at the working point. n is always to be laid off perpendicular to the tangent to the arc at the middle point of the swing. G_1 (see Fig. 34[a]) swings on the arc ED, perpendicular to the axis of X, which is the direction in which the correction is to be applied, bringing this point up to E. For an amplitude m of swing of the radius rod we should therefore have to lay off a smaller distance, giving E' as a point on the desired arc, which distance is best found by construction. In the present case, instead of .15 cm. we lay off .1 cm., and by (2) we finally obtain

$$\rho_1 = 21.5 \text{ cm.}$$

This length was used in constructing Fig. 34, and Fig. 35 gives the resulting valve diagram.

BROWN GEAR NO. I.

In Brown's gear (Fig. 38, Plate 4) rod l_4 takes hold of the eccentric through the intervention of the eccentric rod l, whose upper extremity is guided by a swinging link. By this arrangement the errors inherent in the preceding gears are eliminated. For guiding G_2 Brown ordinarily employs the mechanism shown in Fig. 9, Plate 1. This gear makes a very fair diagram, the errors being well compensated for.

The approximate diagram is obtained as follows:

Lead circle: Dependent on the value of r and on the ratio $\frac{l_5}{l_4}$.

$\xi_1 = r\frac{l_5}{l_4}\cos\omega$, and this valve circle is therefore unchanged for all cut-offs.

$$A = \frac{r}{2} \times \frac{l_5}{l_4}.$$

Cut-off circle: Dependent on the angle of obliquity of the radius arm or the Brown guiding mechanism, and on the ratios

$$\frac{l_4}{l_3} \text{ and } \frac{l_2}{l}.$$

$$\xi_2 = r \times \frac{l_4}{l_3} \times \frac{l_2}{l} \tan a \sin\omega.$$

$$B = f = \frac{r}{2} \times \frac{l_4}{l_3} \times \frac{l_2}{l} \tan a.$$

The *resultant* (ξ) *valve circle* is consequently

$$\xi = r\frac{l_5}{l_4}\cos\omega + r\frac{l_4}{l_3} \times \frac{l_2}{l} \tan a \sin\omega.$$

Errors and their correction.—According to (3), to eliminate the error due to the angularity of l_4,

$$l_6 : l_4 = l_1 : l.$$

This will equalize the vibrations of G_2 to the right and left of its middle position, and we must proportion ρ, the radius arm or its equivalent, so as to neutralize the error due to the valve rod.

We shall have, therefore,

$$l_6 = \frac{l_4 \times l_1}{l} \text{ and } \rho = \frac{S \times l_3}{l_4}.$$

Therefore, to obtain a diagram corresponding to that of the piston with infinite connecting rod, all that it is necessary to fix upon now are the positions of G_2 on l_3 and of G_1 on l_1. We shall, however, endeavor at once to obtain a diagram giving a travel proportional to the piston travel.

$$\text{Assume: } \left.\begin{aligned} l &= 60 \text{ cm.} \\ l_1 &= 20 \text{ cm.} \end{aligned}\right\} \frac{l_1}{l} = \frac{1}{3}$$

$$\left.\begin{aligned} l_2 &= 40 \text{ cm.} \end{aligned}\right\} \frac{l_1}{l_2} = \frac{1}{2}$$

$$\frac{l_5}{l_4} = \frac{1}{3}.$$

$$A = \frac{r}{2}\frac{l_5}{l_4} = \frac{(e+v)}{2}; \quad \therefore r = \frac{(e+v)l_4}{l_5} = 3(e+v).$$

$$\tan a_1 = \frac{e}{2} \times \frac{2 \times l_3 \times l}{3(e+v) \times l_4 \times l_2} = \frac{2e}{3(e+v)}.$$

The length of the valve rod is $112 - (l_1 = 20) = 92$ cm. Hence the angularity $= \frac{m^2}{2\rho} = \frac{[3(e+v)]^2}{2 \times 92} = .44$ cm. To this is to be added the distance .3 cm. to make the diagram correspond with that of the piston, giving $n_1 = .44 + .3 = .75$ cm., about. This applies to G_3; at G_2 we must therefore make a correction $n_2 = \frac{3}{4} \times .75 = .56$ cm.

$$\text{Hence } \rho = \frac{m^2}{2n_2} = \frac{[3(e+v)]^2}{2 \times .56} = 72 \text{ cm.}$$

If we wish to consider the error due to the angularity of l, $n_3 = \frac{2}{3} \times \frac{1}{3} \times \frac{r_2}{2l} = .02$ cm., reduced to point G_3. Subtracting this from .75 gives

$$n_2^1 = (.75 - .02) \times \frac{3}{4} = .55$$

$$\text{and } \rho_1 = \frac{m^2}{2n_2^1} = \frac{[3(e+v)]^2}{2 \times .55} = 74 \text{ cm.}$$

The resulting diagram is shown by Fig. 39, Plate 4. The dimensions chosen are as small as possible, which shows with what degree of accuracy the process followed above gives a desired result even under unfavorable conditions.

Note: In the case just discussed the length of the swinging link l_6 was not determined according to (3), the amplitude of vibration being too great. The error due to l_4 was calculated from the approximate formula (1), and l_6 then calculated from the exact formula: its length was found = 13 cm.

II. VALVE GEARS ACTUATED BY THE CONNECTING ROD.

These gears are operated without aid of an eccentric and are subject to the errors of the connecting rod. The error due to the angularity of a connecting rod of length P for an amplitude of swing R is $n = \frac{R^2}{2P}$. For a point G at a distance l from the wrist-pin this error becomes $n_1 = \frac{R^2}{2P} \times \frac{l}{P}$.

$$\text{For } \frac{l}{P} = \frac{1}{3} \qquad n_1 = \frac{1}{6}\frac{R^2}{P}$$

$$\frac{l}{P} = \frac{1}{2} \qquad n_2 = \frac{1}{4}\frac{R^2}{P}$$

$$\frac{l}{P} = \frac{3}{4} \qquad n_3 = \frac{3}{8}\frac{R^2}{P}$$

In the present case $\frac{R}{P} = \frac{1}{5}$, hence

$$n_1 = \frac{1}{30} R.$$

$$n_2 = \frac{1}{20} R.$$

$$n_3 = \frac{3}{40} R.$$

BROWN GEAR NO. 2, OR JOY GEAR NO. 1.

See Fig. 37, Plate 4.

The *lead circle* is dependent on R and on $\frac{G_2G_3}{G_1G_2} = \frac{l_2}{l}$

$$\xi_1 = R\frac{l_2}{l}\cos\omega \text{ and } A = \frac{1}{2}R\frac{l_2}{l},$$

a constant.

Cut-off circle: Dependent on R, on length $RG_1 = l_1$, on $\tan \alpha$, and $\frac{l + l_2}{l} = \frac{l_3}{l}$, where $l_3 = G_1G_3$.

$$\xi_2 = R\frac{l_1}{P} \times \frac{l_3}{l} \tan \alpha \sin \omega.$$

$$\text{and } B = \frac{R}{2} \times \frac{l_1}{P} \times \frac{l_3}{l} \tan \alpha.$$

$$\tan \alpha = \frac{2B \times P \times l}{R \times l_1 \times l_2}.$$

ξ circle:

$$\xi = R \times \frac{l_2}{l} \cos \omega + R\frac{l_1}{P} \times \frac{l_3}{l} \tan \alpha \sin \omega.$$

$(e + v)$ is therefore $= \frac{Rl_2}{l}$ and $l = \frac{RQ}{R + (e + v)}$, approximately.

Errors.—The path of point G_1 is approximately an ellipse with semi-axes $a = R$, $b = R\frac{l_1}{P}$. The radius of curvature of this ellipse for the points corresponding with the crank at half-stroke is $\rho = \frac{a^2}{b} = \frac{RP}{l_1}$. If the length of l is anywhere near this value, there would be no motion whatever to the left, while that toward the right would be considerable. The error is too great to be compensated for by curvature of the link. In the present case, *e. g.*, for $\frac{l_1}{P} = \frac{1}{2}$, $\rho = 2 \times 20 = 40$ cm. $Q = 44$ cm., hence $l = \frac{20 \times 44}{20 + 3} = 38.3$, almost the same as ρ. We should, therefore, have scarcely any motion toward the left, and double the proper amount toward the right. It is evident that this gear will not give a satisfactory diagram for any length of l applicable to marine engines.

BROWN GEAR NO. 3, OR JOY GEAR NO. 2.

Figs. 40 to 44, Plate 4.

This gear is ordinarily known as the Joy gear. It differs from the one just described by introducing the levers 1 and 2,

Fig. 40, by means of which the errors of that gear are compensated for.

Much depends on the proper location of G_2, and it is ordinarily recommended to find it by trial. It can, however, be fixed in a simple way analytically as follows:

The object is the elimination of the error due to the angularity of rod 3. G_2, as a point in 3, describes an arc of a circle of radius $=\rho = l_3$. If now this point as a point of rod 2 describes a similar arc the error will be eliminated. Consider the connecting rod disconnected from the crank pin and moved vertically. Point G_2 will then, as a point on 2, describe an ellipse with semi-axes G_3G_2 and G_2G_1, which we will call a and b. We shall not go far wrong, as G_2 describes only a small portion of the flat arc of the ellipse, in substituting for this elliptic arc an arc described with the radius of curvature $\rho = \frac{a^2}{b}$. We must therefore make $l_3 = \frac{a^2}{b}$.

Assume now $b = 1$; $a = 2$;

$$l_3 = \frac{a^2}{b} = 4, \text{ or}$$

$$b : a : (a + b) : l_3 = 1 : 2 : 3 : 4.$$

With these proportions the motion of G_1, either side of middle position, is equalized.

With the assumed ratios, the equation to the resultant valve circle is

$$\xi = R \times \frac{2}{3} \times \frac{l_3'}{l_3} \cos \omega + R \times \frac{1}{2} \times \frac{l}{l_3} \tan \alpha \sin \omega.$$

Let us now construct the valve motion In Fig. 41 let G_1 be moved to O:

$$G_2, O = b;\ G_2, G_5 = l_3 \therefore OG_5 = \frac{3}{4} l_3.$$

If now G_1, coinciding with O, be raised a distance R, then O, on rod l_3, will be raised a height

$$h = R \times \frac{2}{3} \times \frac{3}{4} = \frac{R}{2}.$$

Furthermore $OG_5 + G_5G_6 = Q$.

For O_1, $h = \frac{R}{2}$; for G_6, h_1 is to be $= (e + v)$.

From similar triangles

$$\frac{R}{2} : (e + v) = \frac{3}{4} l_3 : Q - \frac{3}{4} l_3$$

$$\therefore \frac{3}{4} l_3 = \frac{RQ}{R + 2(e + v)}.$$

Substituting numerical values ($R = 20$ cm.; $Q = 44$ cm.; $(e + v) = 3$ cm.)

$$\frac{3}{4} l_3 = \frac{20 \times 44}{20 + 6} = 33.84.$$

Hence

$$b = \frac{1}{4} l_3 = 11.28 \text{ cm.}$$

$$a = \frac{1}{2} l_3 = 22.56 \text{ cm.}$$

$$a + b = \frac{3}{4} l_3 = 33.84 \text{ cm.}$$

$$l_3 = 45.12 \text{ cm.}$$

$$l_{3_1} = \left(Q - \frac{3}{4} l_3 \right) = 10.16 \text{ cm.}$$

Errors and their correction.—The angularity of the connecting rod for point G_1 (half way between crank pin and crosshead pin) is $n = \frac{1}{20} R$. For G_2 this is $\frac{2}{3} n = \frac{1}{30} R = .66$ cm.

This would be about the error for G_2 when the crank is at half stroke downward, because for that point the error due to the swinging link 1 may be taken $= 0$. At the corresponding position on the upward stroke we have, however, an error due to the angularity of 1 equal to about .46 cm. $= n_1$, which acts in a direction opposite to the first. For the left side we shall have, therefore, an error of $.66 - .46 = .2$ cm. Taking the mean of these errors to the right and left gives $n_3 = \frac{.66 + .2}{2} = .43$ cm.

This reduced to G_6 gives $n_3 = .43 \times \frac{10.16}{45.12} = .097 = .1$ cm., say. This in conjunction with the error due to the valve rod makes the

length of the radius rod $\rho = 58.9$ or 60 cm. Fig. 42, Plate 4, shows the gear laid down to the above dimensions. The diagram resulting is shown by Fig. 43.

In another form of this gear the valve rod is attached at a point between the radius arm and the connecting rod, and the direction of rotation being unchanged, steam is taken on the inside. For this gear we have assumed $R = 50$ cm.; $\frac{P}{R} = 5$; $Q = 80$ cm.; $S = 125$; $e = 30$ mm.; $v = 15$ mm.

The angularity of the connecting for G_1 is, as before, $n = \frac{1}{20} R = 2.5$ cm. The angularity of the swinging link 1 is 4 cm. toward the right, .5 cm. toward the left; mean value $n_1 = 2.25$ cm.

For G_2 these are

$$\frac{2}{3} n = 1.66 \text{ cm.}$$

$$\frac{1}{3} n_1 = .75 \text{ cm.}$$

They act in opposite directions, their difference therefore, .91 cm., is the quantity to be dealt with.

S in this case is 125 cm.; and $m = 25$ cm.

$$\text{Hence } n_3 = \frac{m^2}{2S} = \frac{25^2}{2 \times 125} = 2.5 \text{ cm.}$$

For $\frac{3}{4} l_3$, as before, $h = \frac{R}{2} = 25$ cm.; $(e + v) = 4.5$ cm. $= h_1$; $Q = 80$; $h = 25$;

$$\frac{3}{4} l_3 = \frac{Q \times h}{(h - h_1)} = 97.56 \text{ cm.};$$

$$\text{and } b = \frac{1}{4} l_3 = 32.52 \text{ cm.}$$

$$a = \frac{1}{2} l_3 = 65.04 \text{ cm.}$$

$$a + b = \frac{3}{4} l_3 = 97.56 \text{ cm.}$$

$$l_3 = 130.08 \text{ cm.}$$

$$l_{3_1} = \left(\frac{3}{4} l - Q\right) = 17.56 \text{ cm.}$$

The error at G_2 (.91 cm.) becomes for G_6

$$n_{G6} = .91 \times \frac{17.56}{130.08} = .12 \text{ cm.}$$

This added to the error due to the valve rod gives, $.12 + 2.5 = 2.62$ cm. This is transferred to G_5 in the ratio of 112.5 (= 130 — 17.5) : 130, for which $n_3 = 2.62 \frac{130}{112.5}$ and $m = 25$ cm. Hence the radius rod $\rho = \frac{m^2}{2n_3} = 102$ cm.

III. VALVE GEARS ACTUATED BY THE CONNECTING ROD AND THE CROSSHEAD.

JOY GEAR.

See Figs. 44 to 48, Plate 5.

This gear makes use of levers 3 and 4, and the beam 2 for correcting the errors of 5. It is of about equal merit with the gears described above. The form represented in Figs. 45 and 47 is for a valve with inside steam lap.

Determination of the valve circles.—

$$F_1G_1 = l; \quad G_2F_1 = l_1; \quad l - l_1 = s$$

Lead circle.—Depends on ratios $\frac{l_1}{l}$ and $\frac{l_3}{l_2}$

$$\xi = R \frac{l_1}{l} \times \frac{l_3}{l_2} \cos \omega$$

$$A = \frac{R}{2} \times \frac{l_1}{l} \times \frac{l_3}{l_2}$$

Cut-off circle.—Depends upon $\frac{p}{P}$, $\frac{l_4}{l_2}$ and $\tan \alpha$.

$$\xi = R \times \frac{p}{P} \times \frac{l_4}{l_2} \tan \alpha \sin \omega$$

$$B = \frac{R}{2} \times \frac{p}{P} \times \frac{l_4}{l_2} \tan \alpha.$$

ξ circle.—

$$\xi = R \times \frac{l_1}{l} \times \frac{l_3}{l_2} \cos \omega + R \times \frac{p}{P} \times \frac{l_4}{l_2} \tan \alpha \sin \omega.$$

Correction of errors.—As a point on rod l_2, G_2 describes the arc of a circle; it remains to be seen what path it will describe as a

member of the system $P-1-2-3-4$. From Fig. 45 we have

$$x = R\left(\frac{l-s}{l}\right)$$

$$x^2 = R^2\left(\frac{l-s}{l}\right)^2 \quad \ldots\ldots\ldots \quad (1)$$

$$y^2 = r^2 = \left(\frac{Rs}{l}\right)^2 \quad \ldots\ldots\ldots \quad (2)$$

From (1) . . . $R^2 = \dfrac{x^2 l^2}{(l-s)^2}$.

From (2) . . . $R^2 = \dfrac{(r^2-y^2)l^2}{s^2}$.

Hence $$\frac{r^2-y^2}{s^2} = \frac{x^2}{(l-s)^2};$$

or $$\frac{x^2}{\frac{r^2}{s^2}\left(l-s\right)^2} = \frac{y^2}{\frac{r^2}{s^2}\times s^2} = 1.$$

G_3 therefore travels on an ellipse whose semi-axes are

$$a = \frac{s}{r}(l-s);\ b = r.$$

Substitute for the elliptic arc its greatest radius of curvature

$$\rho = \frac{\frac{r^2}{s^2}(l-s)^2}{r} = \frac{r(l-s)^2}{s^2}.$$

This is to be the length of l_2.

Let us assume $r = \frac{1}{4}l$; $s = \frac{1}{3}l$.

Taking $r = s$, which would probably be the correct thing, would make l very short, l_2 very long. The ratio $\frac{r}{s} = \frac{3}{4}$ was therefore assumed.

$$\rho = l_2 = \frac{\frac{1}{4}l\left(l-\frac{1}{3}l\right)^2}{\left(\frac{1}{3}l\right)^2} = l.$$

Therefore $l_2 - r = \frac{3}{4}l_2$, and the travel of a point O_1 at $\frac{3}{4}\ l_2$ is,

$h = R \times \frac{2}{3} \times \frac{3}{4} = \frac{1}{2} R$, as for the Brown-Joy gear previously discussed.

As before $l_2 = 45.12$ cm.
and $l = 45.12$ cm.
$s = 15.04$ cm.
$r = 11.28$ cm.

The errors for the points marking the extremities of the vibration of F are formed as follows:

$$n \text{ of the crosshead} = \frac{R}{10} = 2 \text{ cm.}$$

$$n_1 \text{ of } G_2 = 2 \times \frac{2}{3} = \frac{4}{3} \text{ cm.}$$

The swing of rod 3 compensates for about $\frac{1}{3}$ of this error, so that the error at G_3 may be taken equal to that of $G_4 = \frac{R}{20} =$ 1 cm. Correcting for this and for the angularity of the valve rod gives a radius of curvature for the link $= \rho = 66.8$ cm. This length is greater than that of the valve rod, which is 60 cm. A swinging link cannot, therefore, be applied. Joy places G_4 out of the center line of the connecting rod, which, however, is not necessary. The resulting diagram, Fig. 46, gives good results as to cut-off, but shows marked irregularities in the lower valve circle, due to the fact that rods 2 and 4 vibrate in opposite directions, so that rod 3 is not moved equally far from middle position for all corresponding points. The length of the rod G_2G_3 is easily determined from the drawing; in Fig. 45 it is equal to p. Fig. 47 shows the gear laid down from the dimensions found above.

KIRK OR BRICE–DOUGLASS VALVE GEAR.

See Figs. 48 to 53, Plate 5.

This gear, like Joy's, utilizes the motion of a beam, not, however, for correcting errors. A link, 5, vibrates about a point, G_2, of the beam 3, which in its turn vibrates about F. This link receives its motion from a parallel motion, consisting of levers 1, 4, and 7.

DETERMINATION OF THE VALVE CIRCLES.

Lead circle: Depends on R and on ratio $\frac{l_2}{l_1}$.

$$\xi_1 = R \times \frac{l_2}{l_1} \cos \omega.$$

$$A = \frac{R}{2} \times \frac{l_2}{l_1}.$$

Cut-off circle: Depends on R, $\frac{p}{P}$, $\frac{l_3}{l_3}$, $\frac{u}{l_4}$.

$$\xi_2 = R \times \frac{p}{P} \times \frac{l_3}{l_{3_1}} \times \frac{u}{l_4} \sin \omega.$$

$$u = \frac{B \times 2 \times P \times l_{31}}{R \times p \times l_3}.$$

ξ circle:

$$\xi = R \times \frac{l_2}{l_1} \cos \omega + \frac{R \times p \times l_3 \times u}{P \times l_{3_1} \times l_4} \sin \omega.$$

If $\frac{l_3}{l_{3_1}} = 2$, *i. e.*, if the broken lever 1 is made a lever with equal arms, and if $\frac{p}{P} = \frac{1}{4}$, we shall have

$$\xi = R \times \frac{l_2}{l_1} \cos \omega + \frac{R \times u}{2 l_4} \sin \omega.$$

From this equation

$$(e + v) = 2B = R \frac{l_2}{l_1},$$

$$\text{and from this, } l_1 = \frac{RQ}{R + (e + v)}$$

$$l_2 = Q - l_1.$$

Substituting $(e + v) = 3$ cm., $R = 20$ cm., $Q = 44$ cm.,

$$l_1 = \frac{20 \times 44}{23} = 38.3 \text{ cm.}$$

$l_2 = 44 - 38.3 = 5.7$ cm.; *i. e.*, the distance Q is to be divided in this ratio; the actual lengths are determined graphically.

$$u = \frac{4B \times l_4}{R}; \quad B = \frac{e}{2} \text{ (for .5 cut-off).}$$

Hence $u = \frac{2el_4}{R}$.

Assuming $l_4 = 30$ cm., and substituting for e its value $= 27$ mm., we shall have

For .5 cut-off $n_1 = \frac{2.7 \times 2 \times 30}{20} = 8.1$ cm.

For .4 cut-off $n_2 = \frac{2.10 \times 2 \times 30}{20} = 6.3$ cm.

For .3 cut-off $n_3 = \frac{1.60 \times 2 \times 30}{20} = 4.8$ cm.

Errors.—The chief source of error is the form of parallel motion employed by Kirk. The radius rods are made of equal length, and the whole system is turned through an angle $\frac{\beta}{2}$ making the construction inconvenient. Furthermore, this arrangement results in a position of G_3 below the middle position of G_4, giving unequal vibrations of the link. It would be well, therefore, to substitute for the Kirk the ordinary similar form of parallel motion. The lengths of 7 and 4 being the same introduces another error, for this construction would be right only if G_3 had no vertical motion, which is not the case. In Fig. 52 let G_3 be at 1. Let it now be raised to a position corresponding with G_4 in the figure, *i. e.*, a height $(e + v)$ bringing it to G_3 (Valve is supposed to take steam on inside). Then rod l_5 will be too long by n_1, and must be drawn back this amount. In this figure we have approximately

$$(e + v) : n_1 = l_5 \text{ (mean value)} : R.$$

$$l_5 = Q.$$

$$\text{Hence } n_1 = \frac{(e + v)R}{l_5} = \frac{(e + v)R}{Q}.$$

G_4 is therefore to travel on the arc of a circle whose n is smaller by n_1 than that of l_5.

By (2) ρ (for inside steam lap) $= \frac{m^2\rho}{m^2 - 2\rho n_1}$

$$= \frac{R^2Q}{R^2 - 2Q\frac{(e + v)R}{Q}} = \frac{RQ}{R - 2(e + v)}.$$

G_4 is to move in the arc of a circle of which this is the radius.

This is accomplished by making rod 7 of this length. The calculation just made was for inside steam lap, for which Kirk always designs his gear to get a better distribution. For outside lap, where $v = -v$,

$$\rho_1 = \frac{RQ}{R + 2(e + v)}.$$

The resulting diagrams are fair for inside, but not very satisfactory for outside steam lap. For this reason it is best to construct the gear using Angström's parallel motion. (See Fig. 14, Plate 4.)

In this case we must, however, make a correction for the error due to the angularity of the beam. Let this error for l_2 be n. From Fig. 51, Plate 5,

l_5 (inside lap) $= \sqrt{[R - (e + v)]^2 + Q^2}$.

l_5 (outside lap) $= \sqrt{[R + (e + v)]^2 + Q^2}$.

Then $\rho = l_7 = \dfrac{R^2}{2(l_5 - Q \mp n)}$, using $-n$ for outside, $+n$ for inside lap.

The diagrams are satisfactory, although the lengths are rather short for obtaining the best results. It is to be recommended, however, to use only inside steam lap, as for outside lap the gear becomes too cramped. See Fig. 49, Plate 5.

Plate 1.

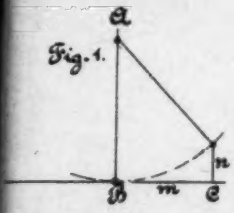


Fig. 1.
A
B
m
C
n

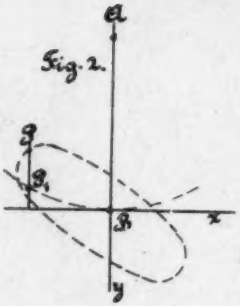


Fig. 2.
A
B
x
y

Fig. 4.
A
m

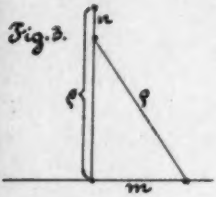


Fig. 3.
n
m

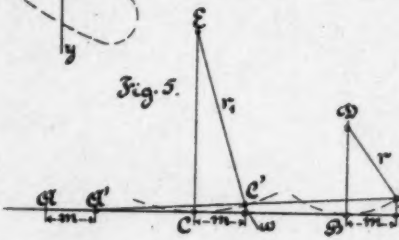


Fig. 5.
E
D
A
A'
C'
C
B

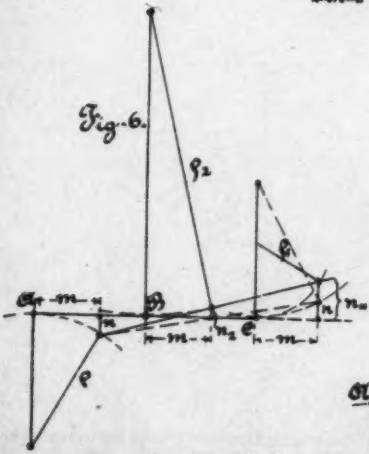


Fig. 6.
A
B
C

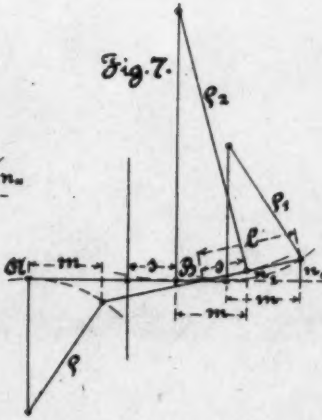


Fig. 7.
A
B

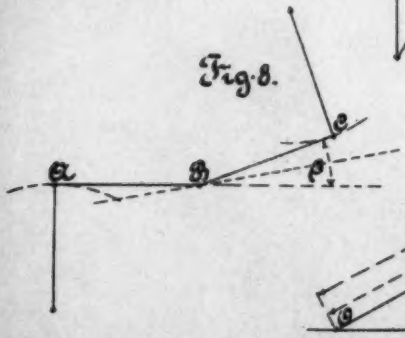
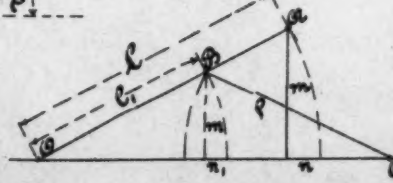
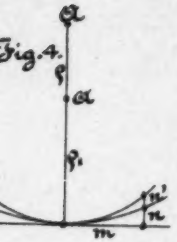
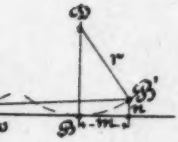


Fig. 8.
A
B
C


A
B
C
m
n





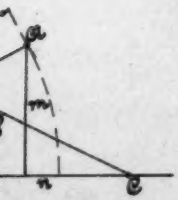
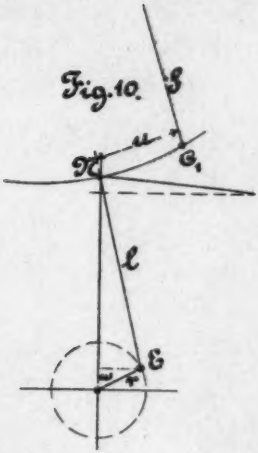


Fig. 10.



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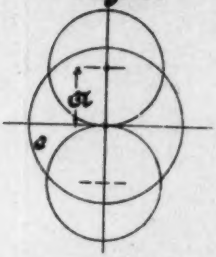
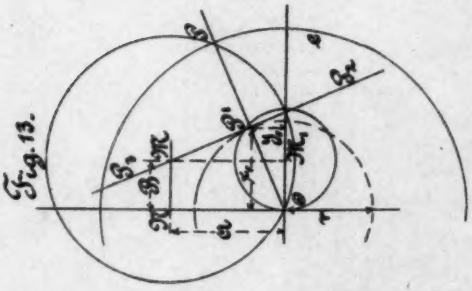


Fig. 12.

Fig. 13.



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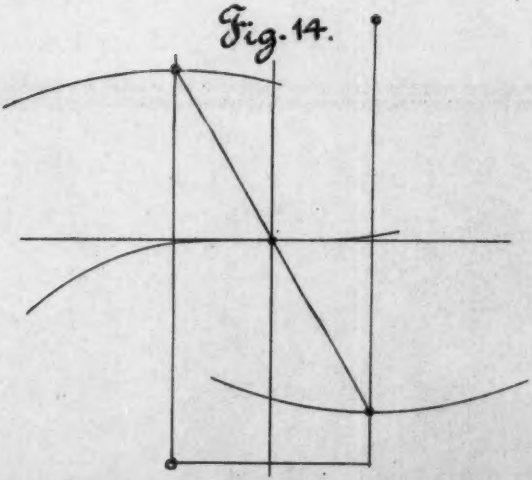

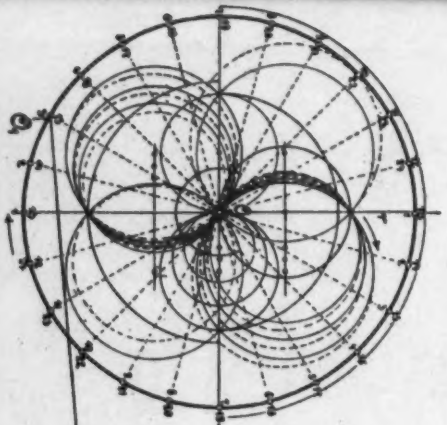

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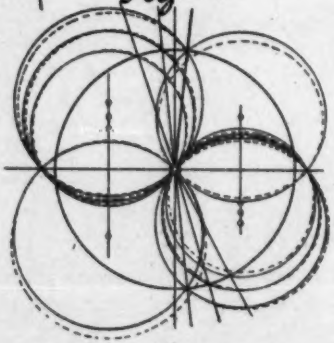


Inside steam laps.

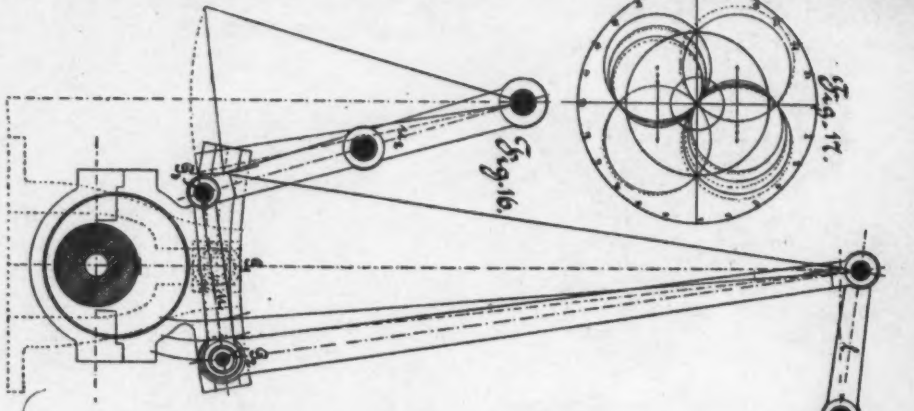


Fig. 16.

Fig. 17.

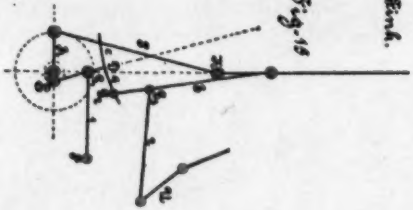
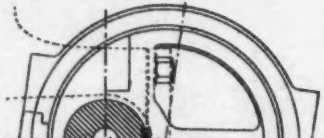


Fig. 18

Link.



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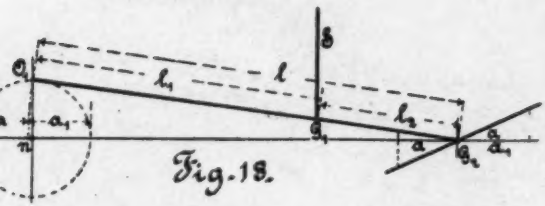

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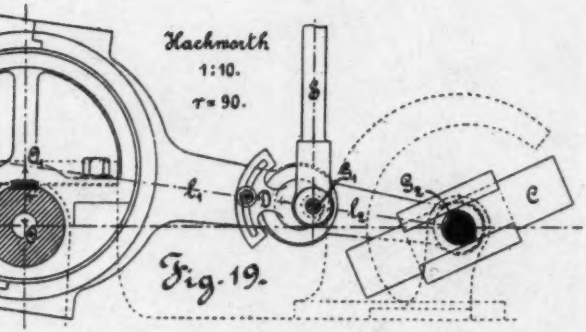
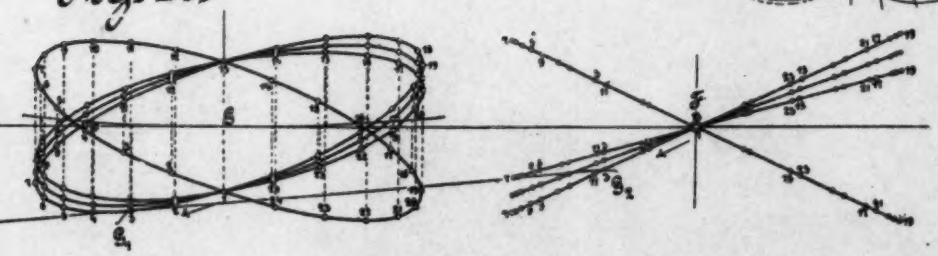


Fig. 19.

Fig. 20.



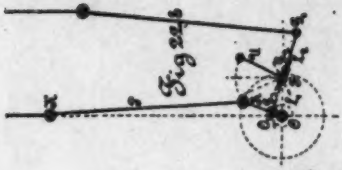


Fig. 22b

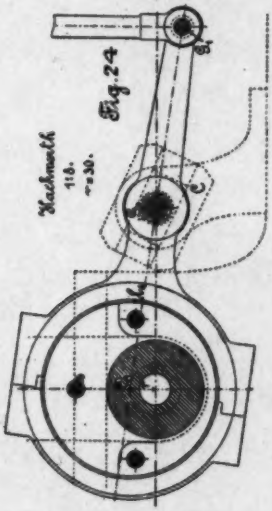


Fig. 24

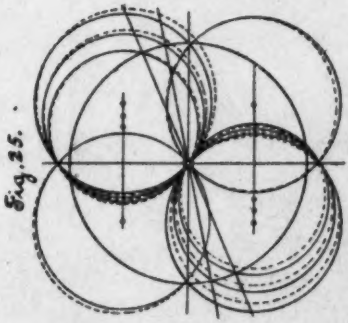
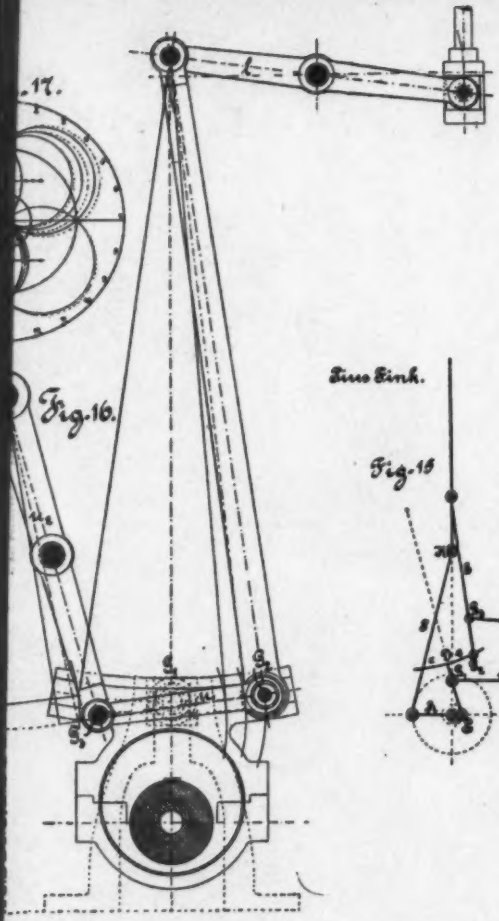


Fig. 25.

Fig. 23.

Plat



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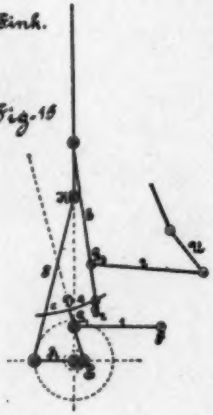
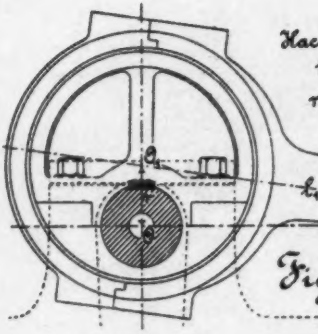


Fig. 15



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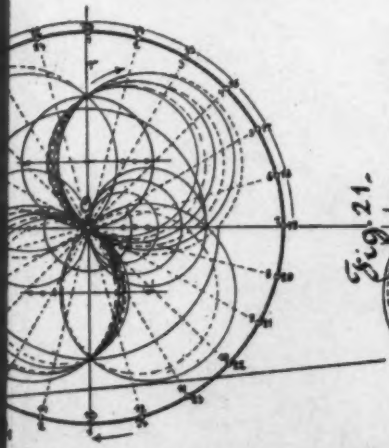

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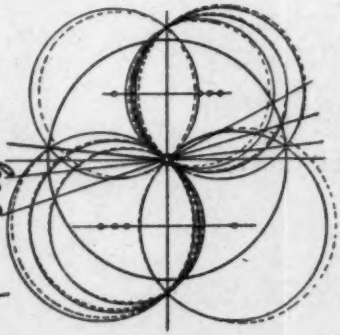
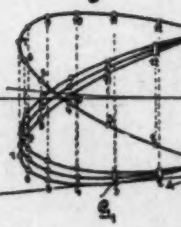


Fig. 20

Inside steam cam.



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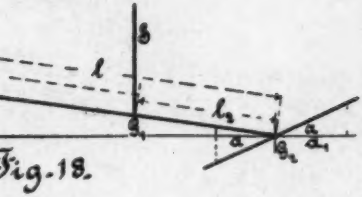

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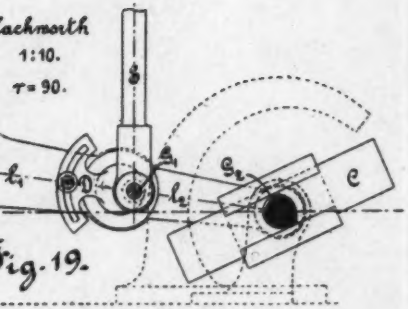

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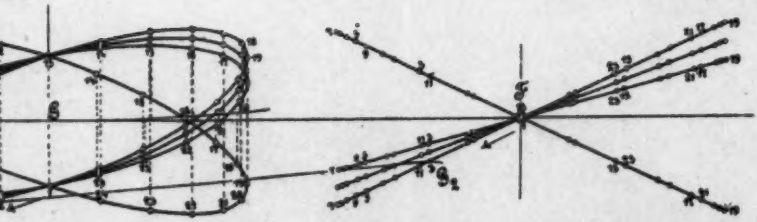

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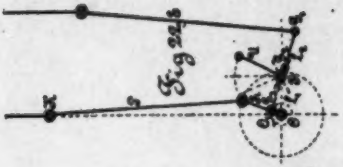

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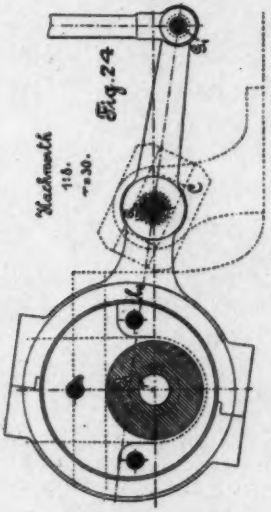


Fig. 24.

Fig. 23.

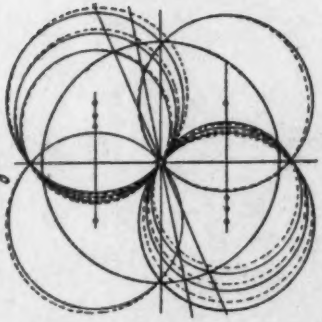
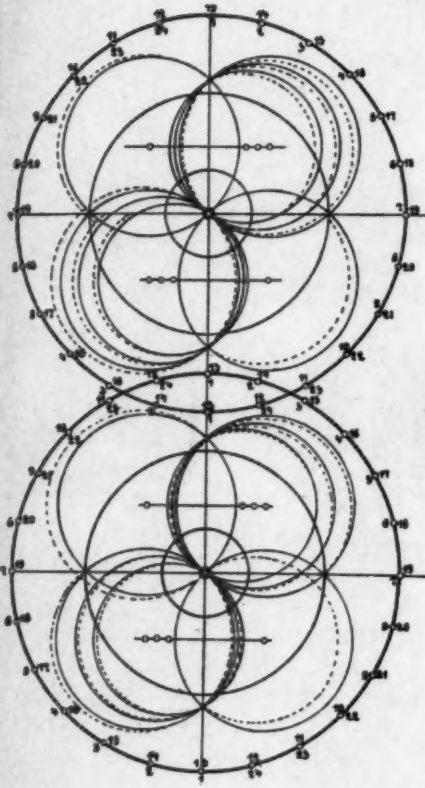


Fig. 25.

Plate 3.



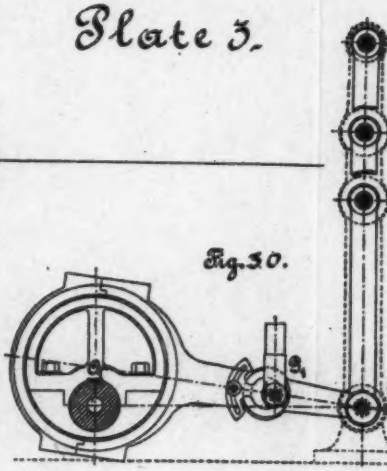


Fig. 30.

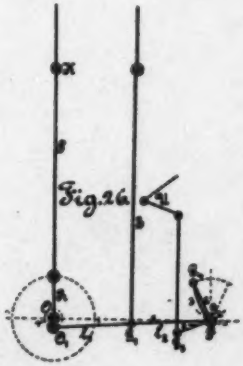


Fig. 24.

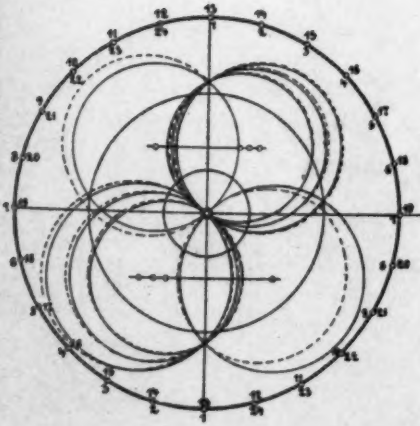
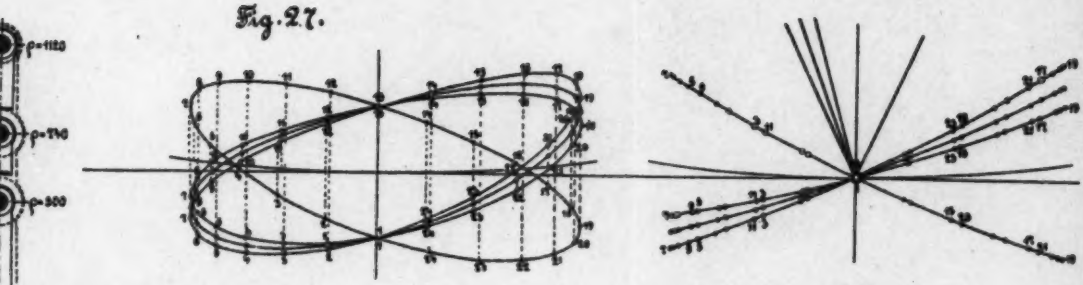


Fig. 27.



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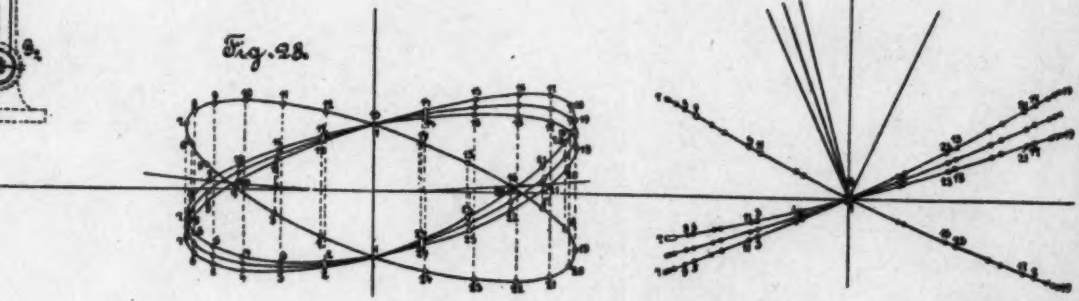
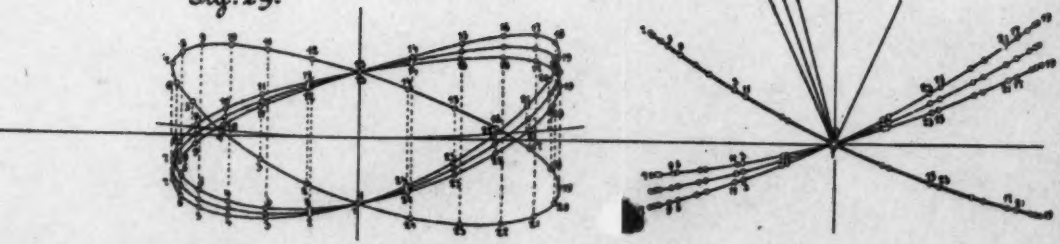


Fig. 29.



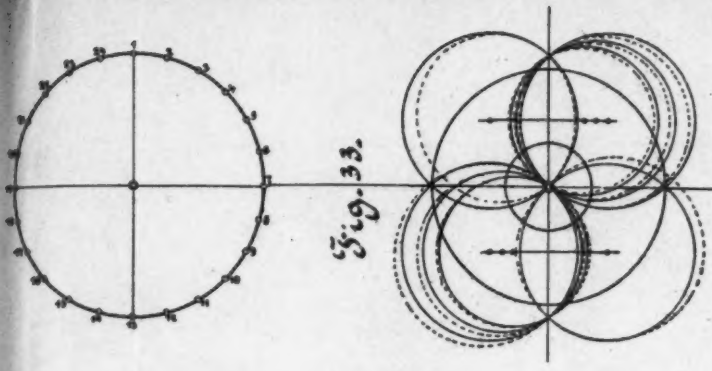
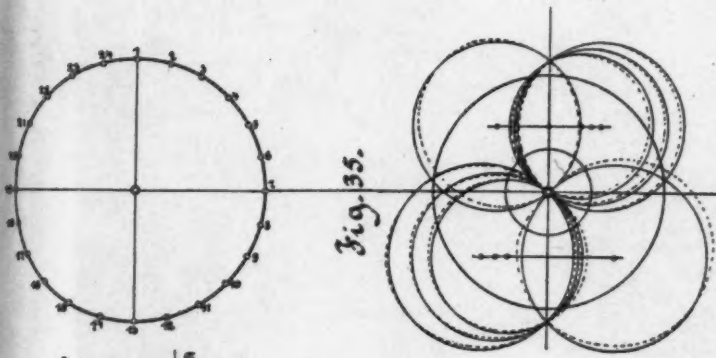


Fig. 33.



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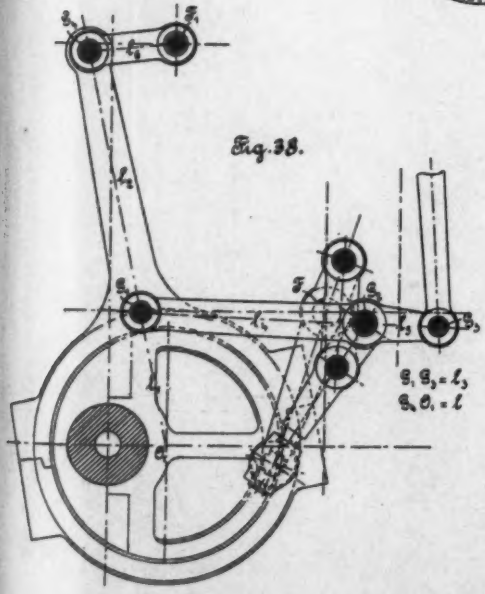

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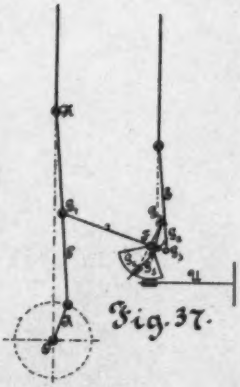


Fig. 37.

Plate 4.

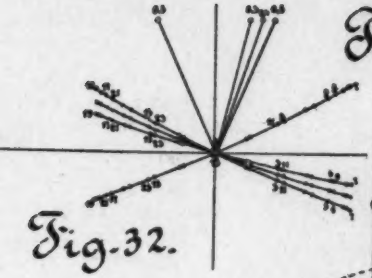
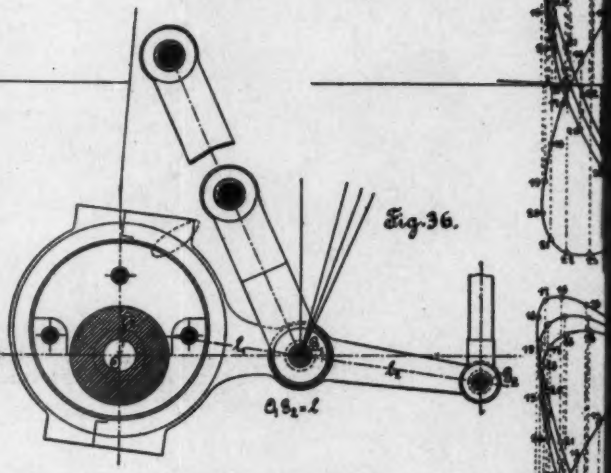


Fig. 32.

Fig. 31.



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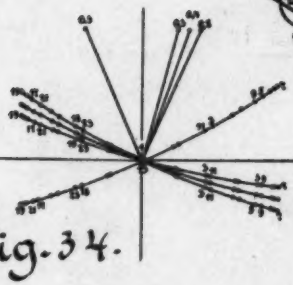


Fig. 34.

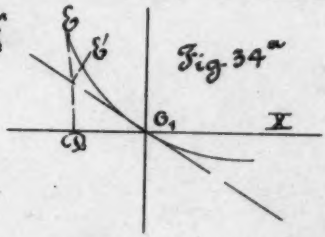


Fig. 34a

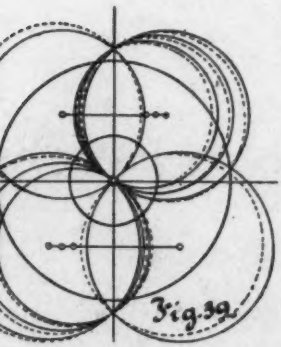


Fig. 39.

Braun. (Joy.)

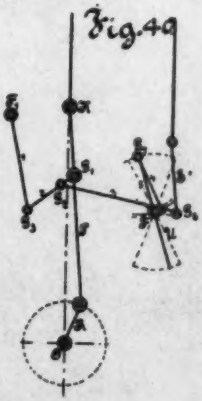


Fig. 40

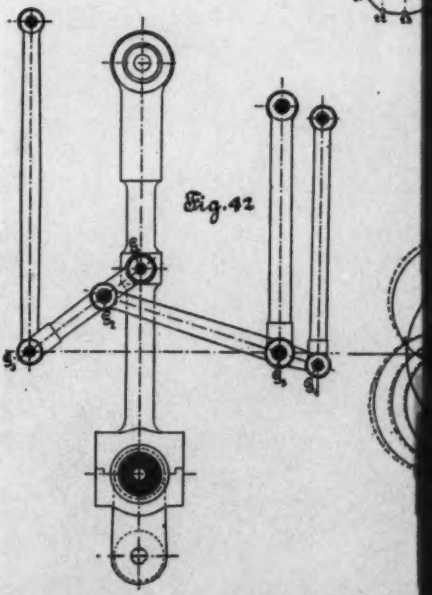


Fig. 42

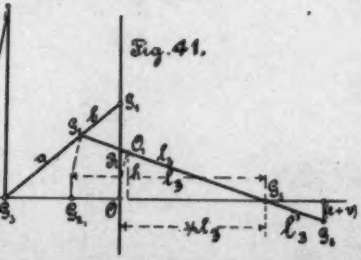


Fig. 41.

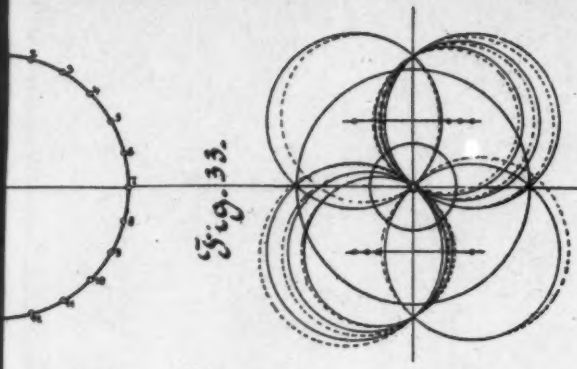
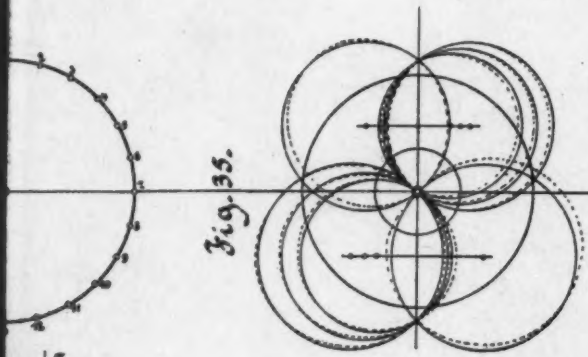


Fig. 33.

Fig. 32.



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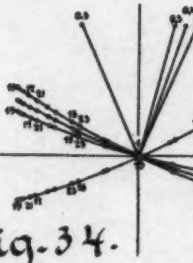


Fig. 34.

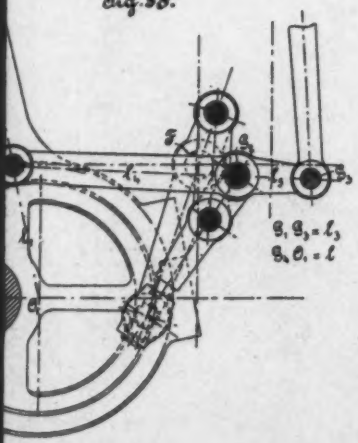


Fig. 38.

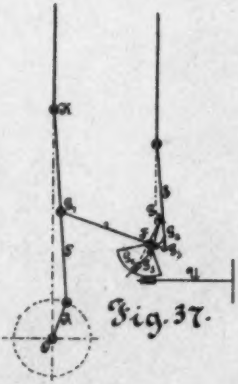


Fig. 37.

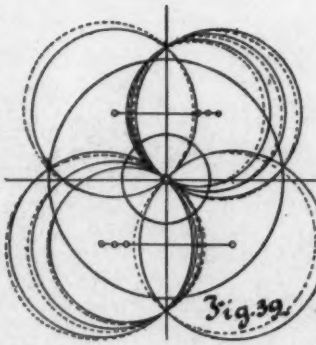


Fig. 39.

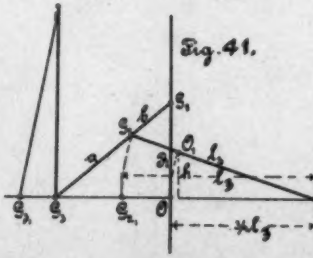
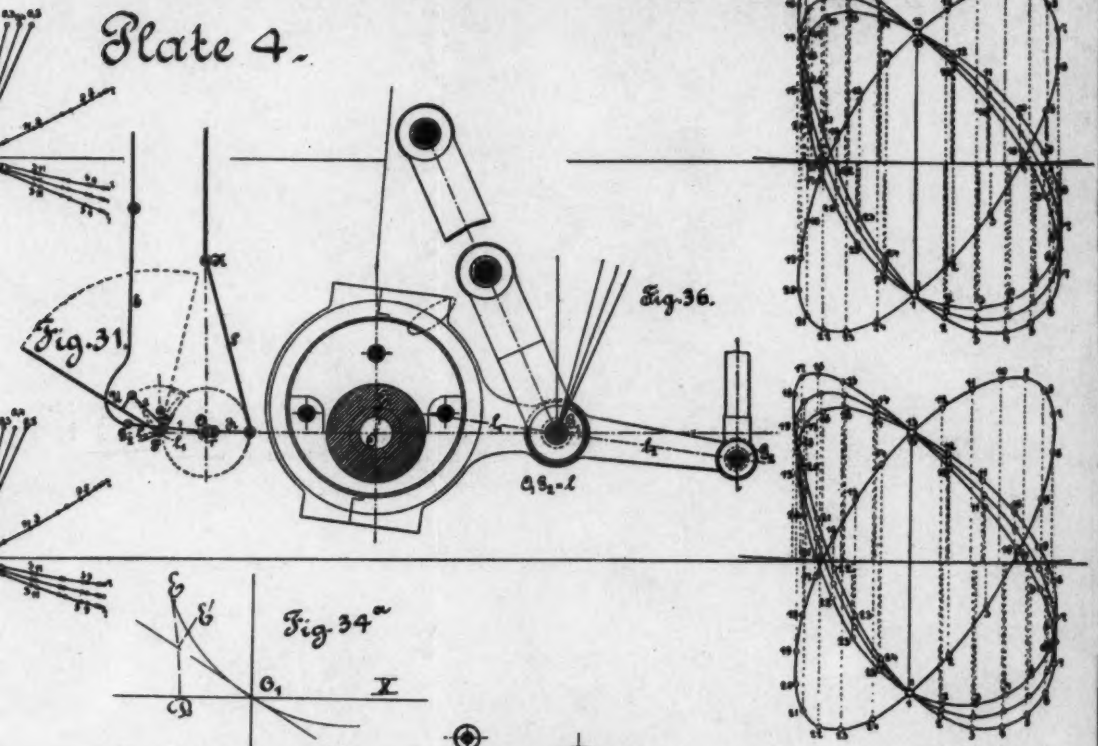
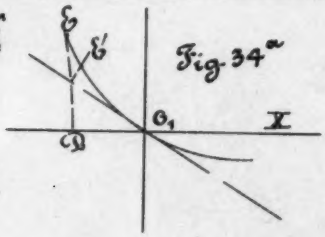


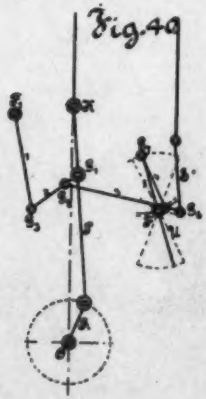
Fig. 41.

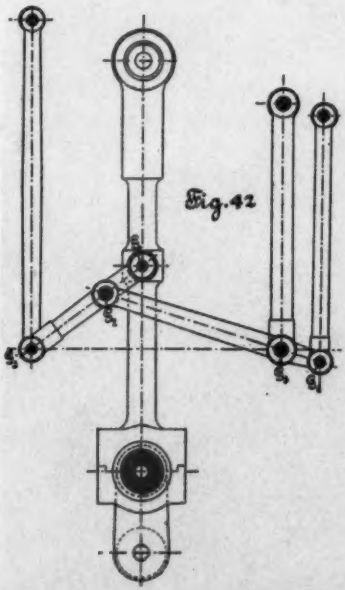
Plate 4.

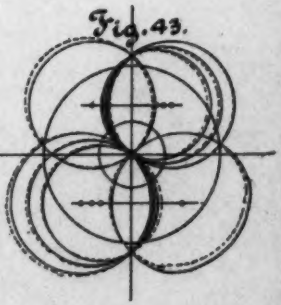




Brauer. (Joy.)







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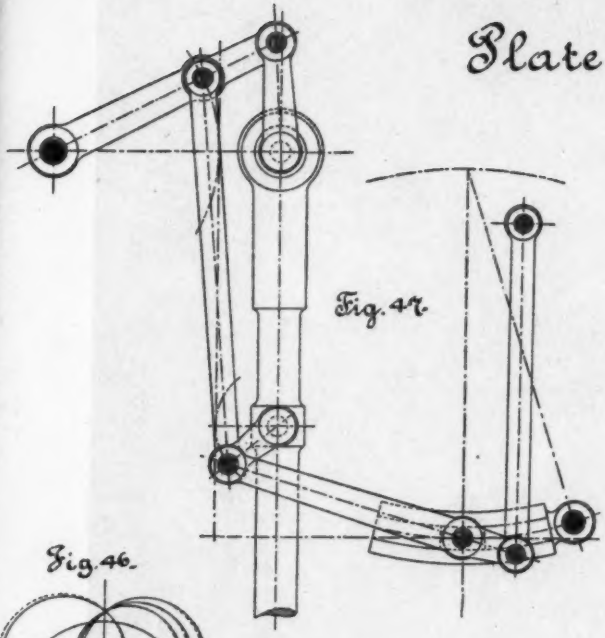
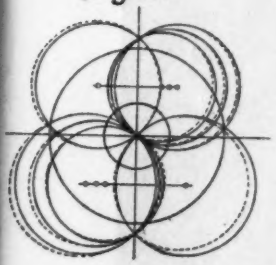


Fig. 47.

Fig. 46.



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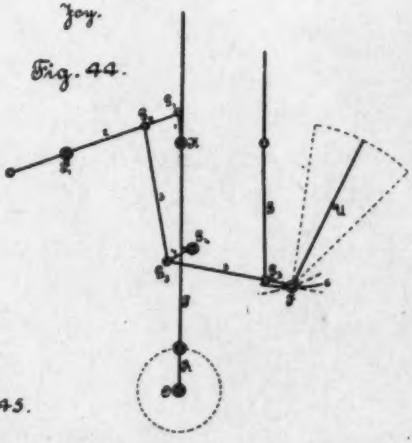


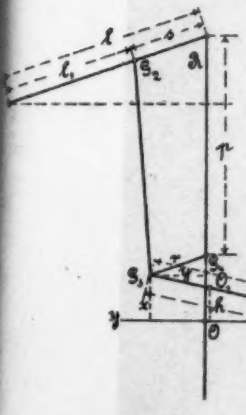


Fig. 45.

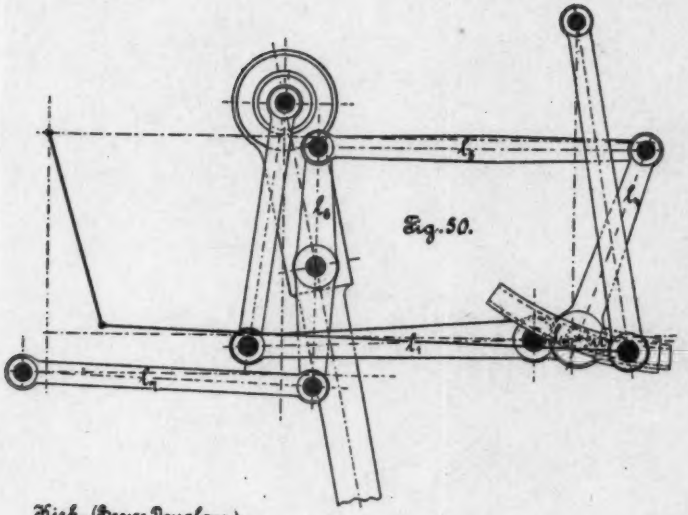
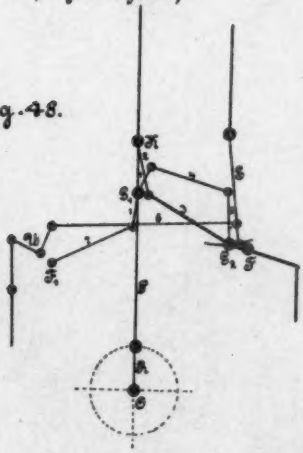


Fig. 50.

Kirk. (Bryce Douglass.)

Fig. 48.



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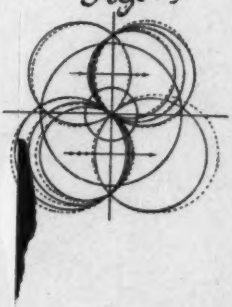
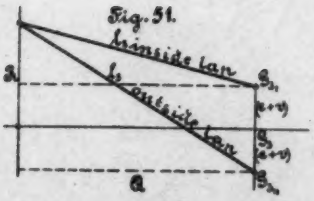


Fig. 51.



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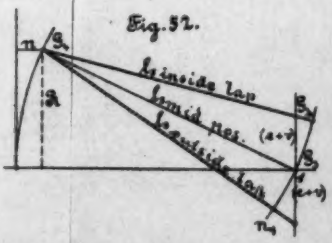

XXXIII.

SPEED CURVES OF SHIPS BUILDING FOR THE U. S. NAVY.

The following speed curves are intended to show the probable performance of the various ships now building for the Navy, by giving the performance of similar ships reduced to a common displacement by Froude's Law of Comparisons. The calculations were made and the curves constructed by Assistant Engineer C. A. Carr, U. S. Navy, while on duty at the Bureau of Steam Engineering. The curves published in the present number are, with the exception of those for the cruisers *Cincinnati* and *Raleigh*—which were designed to make 20 knots—those of ships in whose case the requirement is the development of a certain I.H.P. on trial. Only the machinery of the *Maine*, and the *Monterey* complete, are being built by contractors; the others are building at Navy Yards. The curves of the remaining ships—all being built by private firms—in whose case a certain speed on trial is required by contract, will be published as the trials are reported, and the actual results can then be plotted, and comparison made with the estimate.

According to Froude's Law, if two ships have the same coefficient of fineness, same water lines, etc., but have dimensions whose ratio is n, then for speeds S_1, S_2, S_3, of one, with resistances R_1, R_2, R_3, the corresponding speeds will be $S_1\sqrt{n}$, $S_2\sqrt{n}$, $S_3\sqrt{n}$, for resistances n^3R_1, n^3R_2, n^3R_3. In similar ships the cube of the ratio of dimensions is equal to the ratio of displacements, or n^3 = ratio of displacements, and $\sqrt{n}$ = sixth root of the same ratio.

Let R, S, D, and P represent the resistance, speed, displace-

ment and indicated horse power for the large ship, and r, s, d, and p the corresponding quantities for the small one. Then

$$S = s\sqrt[6]{\frac{D}{d}}. \qquad R = r\frac{D}{d}.$$

Considering $P = k \times RS$, we have

$$P = krs\sqrt[6]{\frac{D}{d}} \times \frac{D}{d},$$

$$P = p\sqrt[6]{\frac{D}{d}} \times \frac{D}{d}.$$

As an example, take the *San Francisco;* bring her up to 5,500 tons, the displacement of *Cruiser No. 6*, and find the corresponding speed and power.

D = 5,500	log 3.74036		
d = 4,088	log 3.61151		
$n^3 = \frac{D}{d}$	log .12885		
$\sqrt[3]{n} = \sqrt[6]{\frac{D}{d}}$	log .02148	log .02148	
p = 9,718	log 3.98758		
s = 19.52		log 1.29048	
P = 13,737	log 4.14791		
S = 20.51		log 1.31196	

The data used in computing the points of these curves has been taken from the records of progressive trials wherever available, and from reliable reports of official trials.

The following table gives the principal dimensions of the vessels used in computing the points of the curves:

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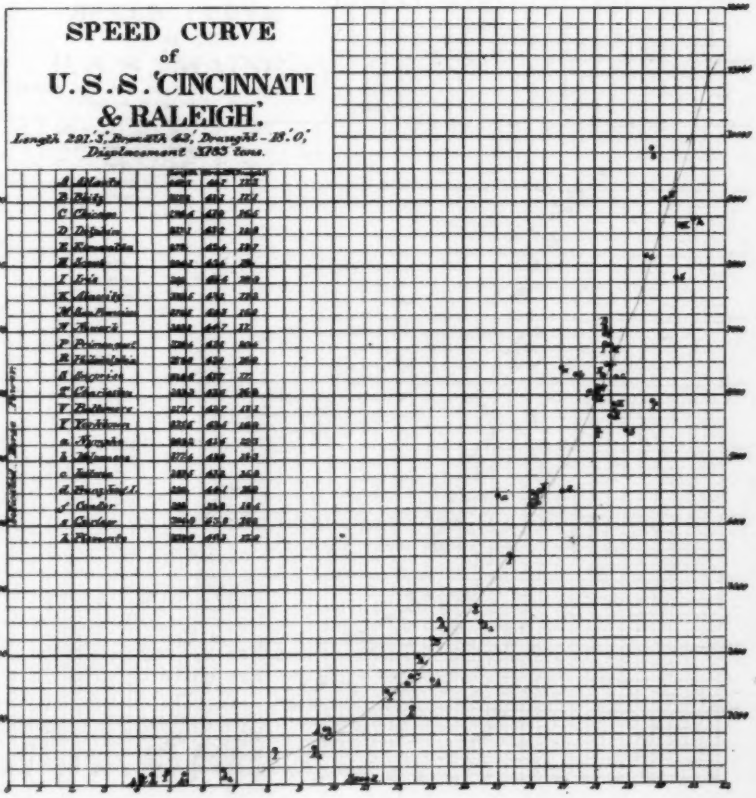
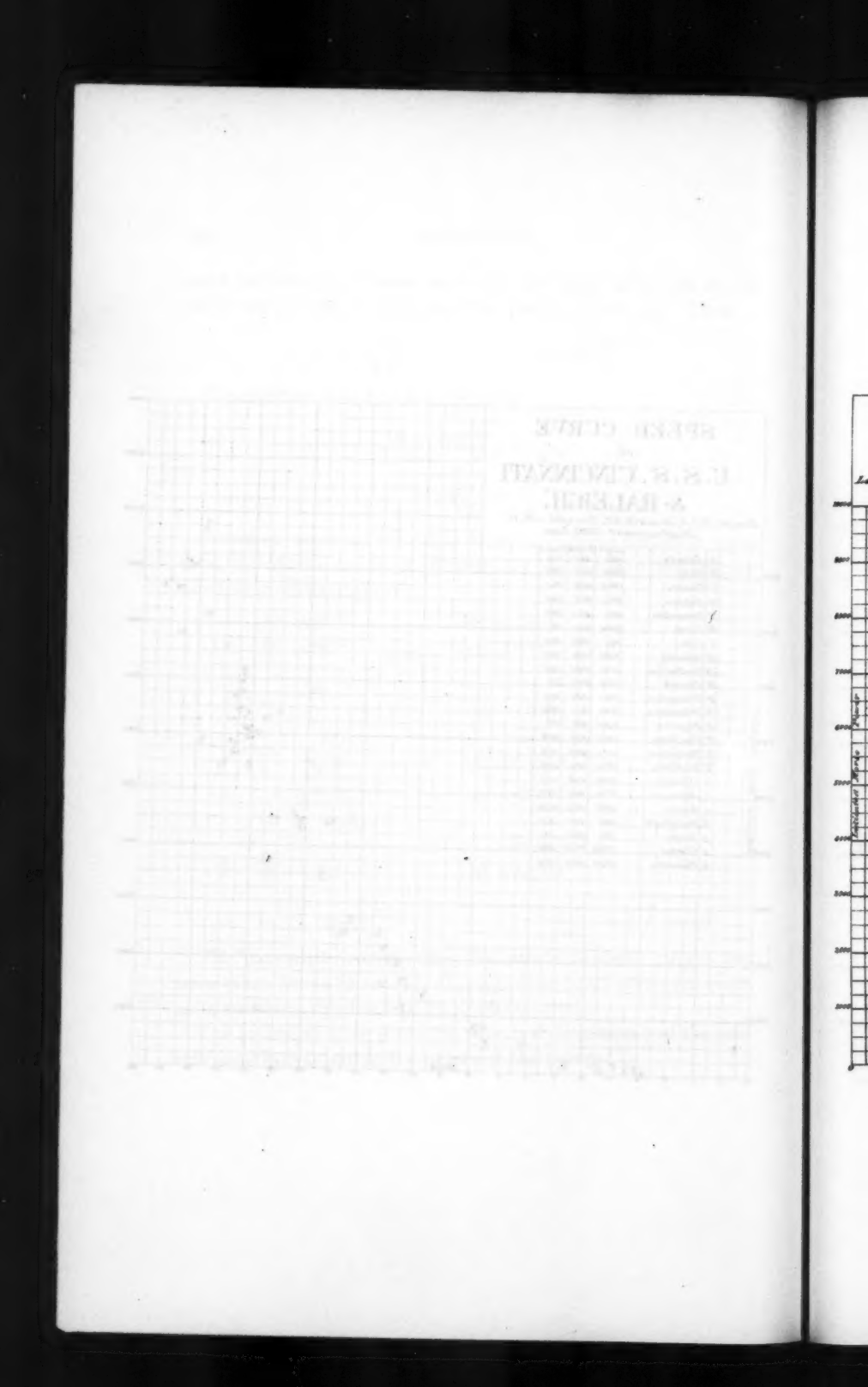
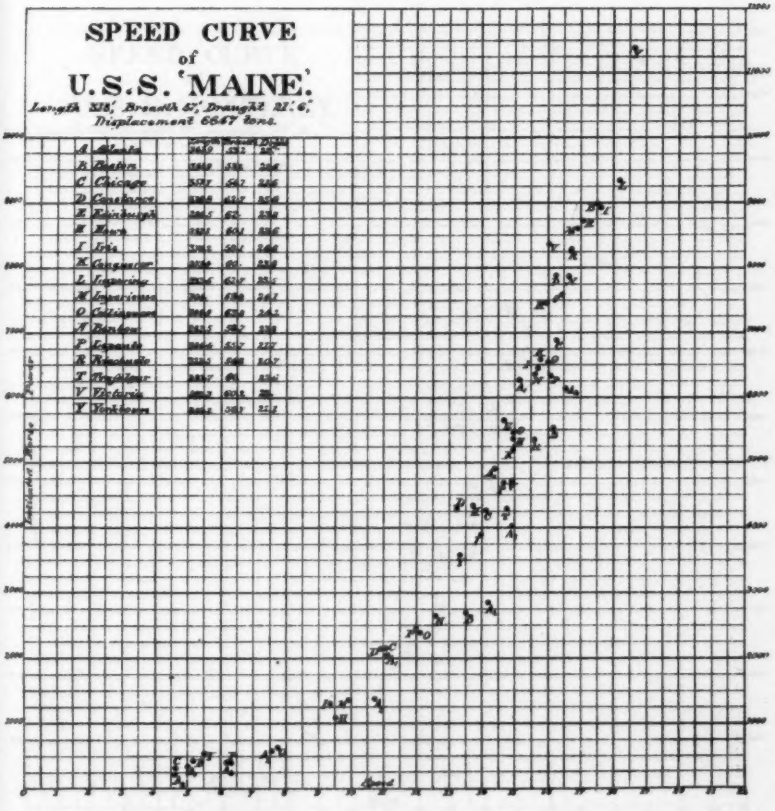
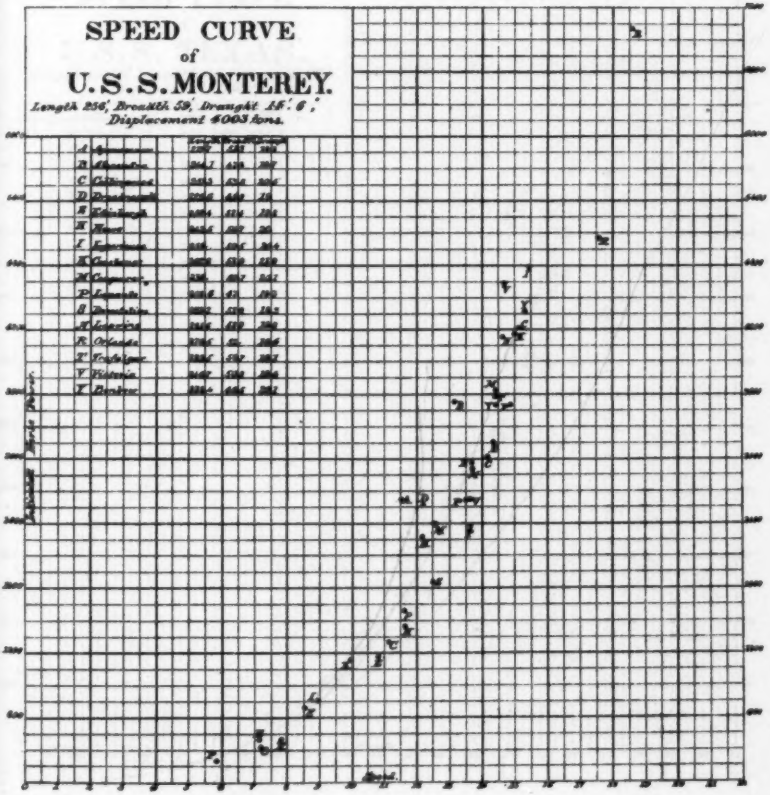

SPEED CURVE
of
U.S.S. 'CINCINNATI
& RALEIGH.'
Length 291'.5', Breadth 42', Draught - 18'.0',
Displacement 3183 tons.

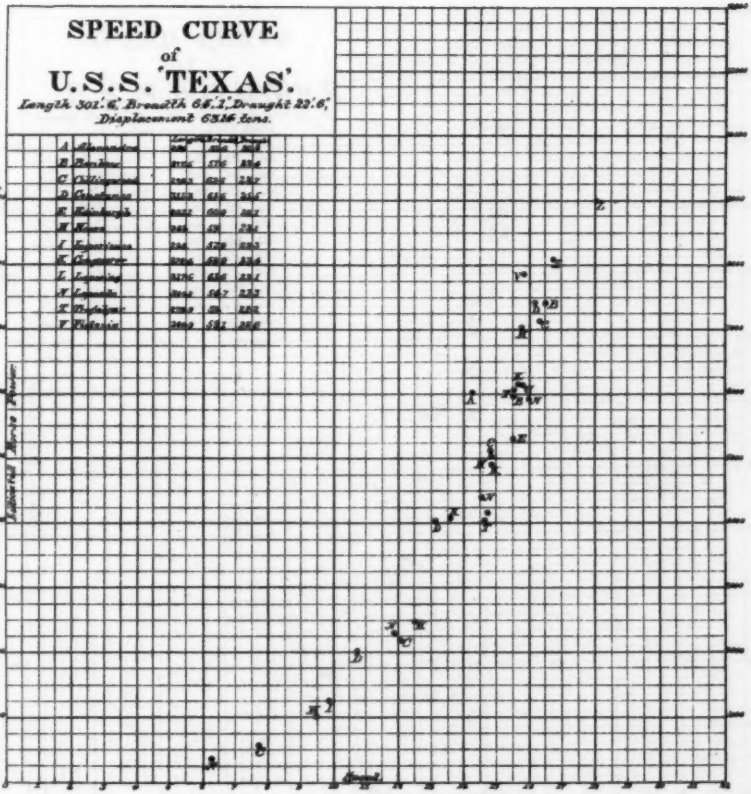


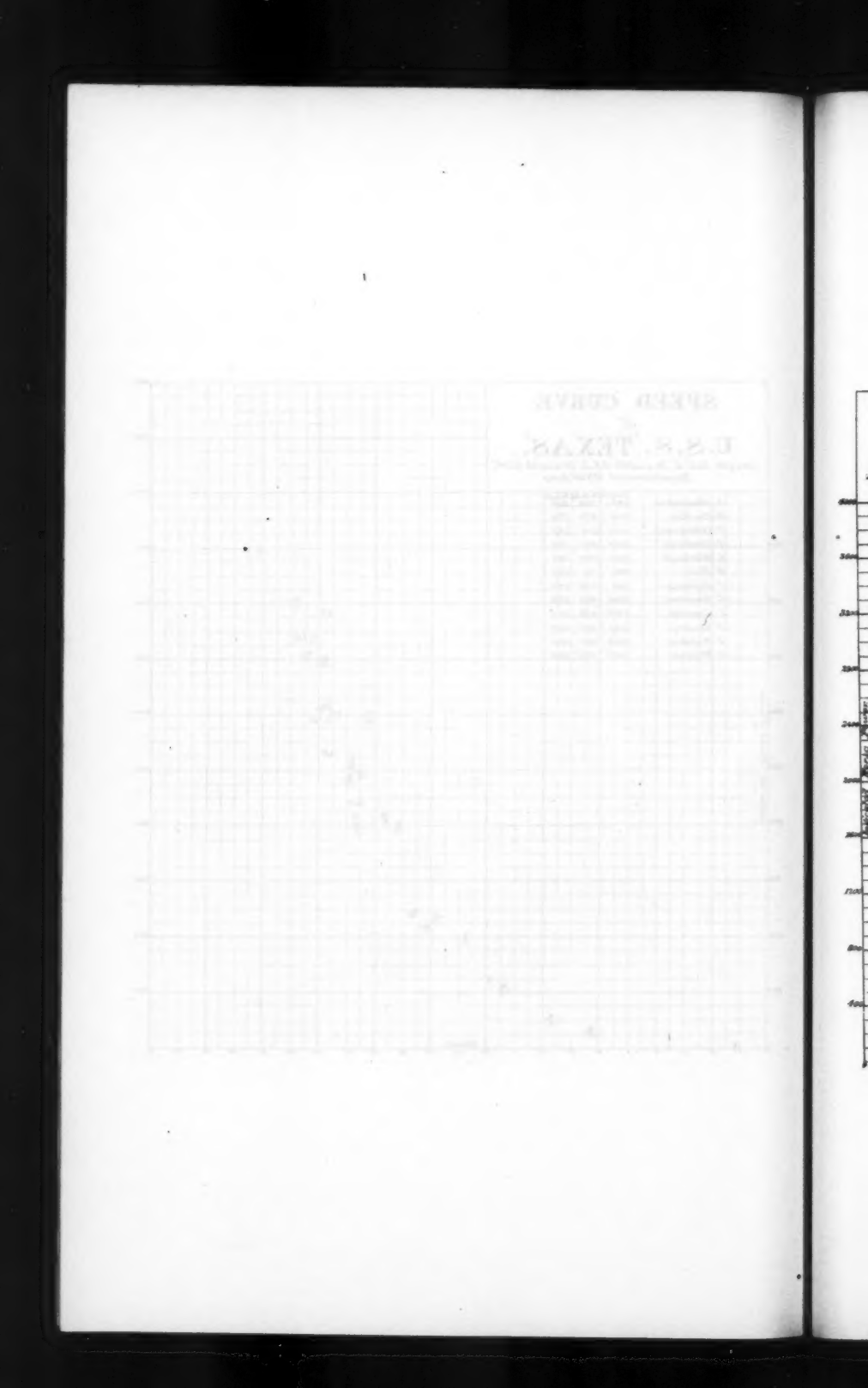

SPEED CURVE
of
U.S.S. 'MAINE.'
Length 318', Breadth 57', Draught 21'. 6".
Displacement 6647 tons.
A Alliance
B Boston
C Chicago
D Constance
E Fantome
H Hawk
I Iris
K Conqueror
L Impeuse
M Imperieuse
O Calliope
T Tauton
P Leander
R Rinaldo
T Trafalgar
V Victoria
Y Yorktown
Speed

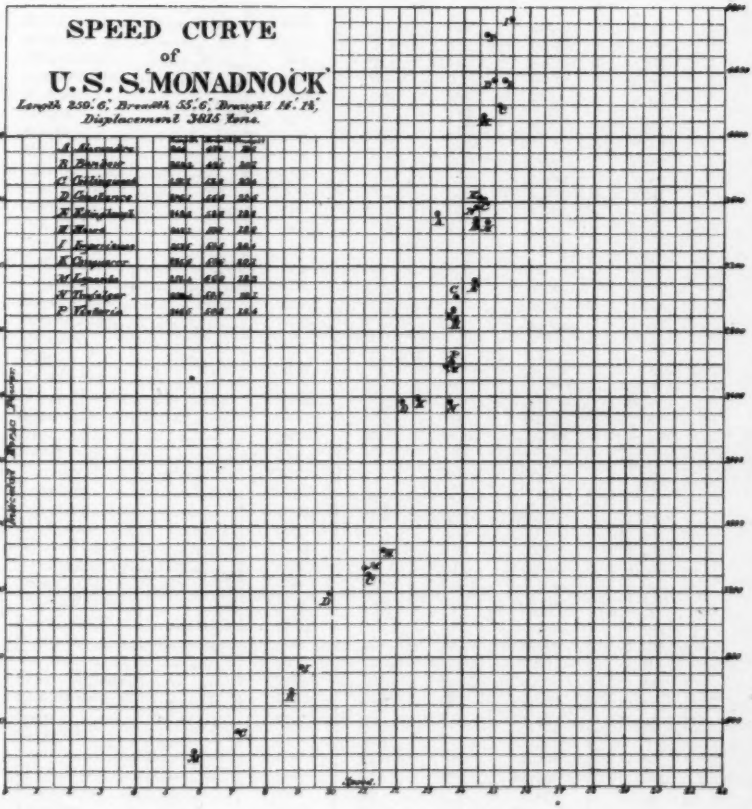
SPEED CURVE
of
U.S.S. MONTEREY.

Length 256, Breadth 59, Draught 15′. 6″, Displacement 6003 tons.




SPEED CURVE
of
U.S.S. 'TEXAS'.
Length 301'.6" Breadth 64'.1" Draught 23'.6"
Displacement 6316 tons.




SPEED CURVE
of
U. S. S. MONADNOCK
Length 259. 6". Breadth 55. 6". Draught 14. 1".
Displacement 3815 Tons.

PRINCIPAL DIMENSIONS OF SHIPS.

Name.	Length between perpendiculars.	Beam.	Mean draught.	Displacement on trial.	Ratio of length to beam.
Atlanta	270.	42.	17 3/8	3,275.	6 43
Alacrity	250.	32 1/2	13.	1,274.	7.70
Alexandra	325.	63 3/4	26 1/2	9,432.	5.10
Agamemnon	280.	66.	23 1/2	8,510.	4 24
Baltimore	315.	48 1/2	19 1/2	4.391.	6 49
Boston	270.	42.	17 3/8	3,275.	6 43
Benbow	330.	68 1/2	27 5/8	10,600.	4 82
Blitz	245.	32 7/12	13 1/4	1,382.	7.52
Chicago	315.	48 1/2	19	4.543.	6 54
Charleston	300	46 1/2	17 5/8	3.557.	6 49
Constance	225.	44 1/2	18 9/12	2,380.	5 00
Condor	216 5/12	29 1/6	14	1,272.	7 42
Curlew	195	28.	10 5/12	785.	6.96
Collingwood	325.	68.	25 1/4	8,060.	4.78
Conqueror	270	58.	23	6 000.	4 65
Dreadnought	320.	63 5/6	26 1/2	10,820.	5 01
Devastation	311 1/2	71.	24 1/2	9,639.	4 38
Dolphin	240.	32.	14 1/2	1,300.	7.50
Esmeralda	270.	42.	18 1/2	2,900.	6.43
Edinburgh	325.	68.	25 3/4	8.760.	4.78
Franz Josef I	321 1/2	48 7/12	18 7/12	3 990.	6.61
Howe	325.	68.	26 3/4	9.637.	4 78
Iris	300.	46.	19 1/2	3,290.	6 52
Imperieuse	315.	62.	25.	7.390.	5 16
Latona	300.	43.	16 1/2	3,400.	6 98
Lapwing	165	31	11 5/8	805.	5 32
Lepanto	400 1/2	72 2/4	28 1/2	14,860.	5 51
Melpomene	265.	42	17 1/2	2.960.	6 31
Newark	310 3/8	49 1/8	18 3/4	3 980.	6.31
Nymphe	195.	30.	12 1/2	1,140.	6 50
Orlando	300	56	20 1/12	5,000.	5.31
Philadelphia	315.	48 1/2	19 1/6	4.325.	6.49
Primauguet	280 3/8	37 5/12	18 1/2	2,236.	7.65
Piemonte	300.	38.	15	2,500.	7 90
Riachuelo	305	52	19 7/8	5,677.	5 86
Reina Regente	306.	50 7/12	19 1/12	4.950.	6 04
San Francisco	300.	49 3/8	18 3/4	4.088.	6 31
Scout	220	34	13 1/2	1,285.	6.42
Surprise	250.	32 1/2	13.	1.320.	7 70
Swatara	216.	37.	16 1/2	1.900.	5 84
Trafalgar	345.	73	27 1/2	11.940.	4 72
Vandalia	216.	39.	17 1/2	2,100	5 54
Victoria	340.	70	26 1/4	10.470	4 86
Yorktown	228.	36.	14.	1.701.	6.33

XXXIV.

THE SELECTION OF THE MOST SUITABLE FORM OF VANE FOR CENTIFUGAL PUMPS.

By Prof. J. Bartl.

[Translated from the Zeitschrift des Vereines Deutscher Ingenieure, of Sept. 12, 1891.]

The following translation is interesting not only on account of its own merit, but also as being a companion piece to the articles on the same subject by Assistant Engineer H. G. Leopold, U. S. N.:

The vanes of centrifugal pumps may, to satisfy varying conditions, be given a variety of forms. The form in ordinary use gives a vane curved backwards; but vanes curved forward, or in the direction of motion of the wheel, are also found. Their advantage consists in the low velocity of the wheel they require for a given flow and head.

In what follows, the peculiarities of the different shapes will be examined, as far as this is possible, on a purely theoretical basis, and the various forms will be compared as to velocity of rotation required, facility of starting, pressure relations within the pump, hydraulic resistances, and leakage.

General considerations.—The theory of centrifugal pumps is best based on the principle of turbine action, according to which the pressure due to 1 kg. of water passing through the machine is expressed by

$$h_n = \frac{1}{g}\,(v_2 c_{u2} - v_1 c_{u1}) \qquad (1)$$

c_2, w_2, v_2 (see Fig. 1) are the absolute and relative velocity of the water and the velocity of rotation of the wheel at *II*. c_{u2} and c_{r2} are the tangential and radial components of the absolute velocity c_2. Letters c_1, w_1, v_1 and c_{u1} and c_{r1} have similar signification for the point of efflux *I* (for a turbine). h_n is the height through

which the water passing through the turbine would have to fall to perform the work transmitted to the vanes on its passage through the wheel.

Reversing the motion of the wheel and the water, and keeping the absolute value of the velocities the same, we shall have the work performed for one kg. of water,

$$H_u = \frac{1}{g}(v_1 c_{u1} - v_2 c_{u2}) = -h_u,$$

which is equivalent to saying that in the second case the wheel communicates to the water energy sufficient to lift it to the ideal height

$$h_u = \frac{1}{g}(v_2 c_{u2} - v_1 c_{u1}).$$

This is the case of the centrifugal pump.

In the case of the turbine the water issues, generally speaking, in a normal—in the present case radial—direction. Likewise, in the absence of guide-vanes the direction of the entering water in a centrifugal pump may be taken as radial; hence $c_{u1} = 0$ and the head due to the wheel becomes

$$h_u = \frac{1}{g} v_2 c_{u2} \qquad (2)$$

Calling the actual head against which the water is delivered h, the discharge velocity c_o, and the sum total of all hydraulic resistances within the pump and the pipes z, then we shall have for the total head h_u

$$h_u = h + z + \frac{c_o^2}{2g} = \zeta h = \frac{1}{g} v_2 c_{u2} \qquad (3)$$

The value of ζ may be assumed as 1.4 to 1.6, provided the water enters without shock, and is conducted from the wheel to the delivery pipe through a gradually increasing conduit. c_1 is usually taken between 1.0 and 2.5 m. per sec.; the ratio

$$\frac{r_2}{r_1} = 1.5 \text{ to } 3.5.$$

In what immediately follows, the form of vane corresponding to any assumed velocity of rotation is obtained graphically. It

is assumed that 16 ltrs. of water per sec. are to be lifted to a height of 10 m.; $\zeta = 1.5$, $c_1 = 2.0$ and $\frac{r_2}{r_1} = 2.5$. In Fig. 3 the vanes are drawn $\frac{1}{2}$ natural size, while the velocities are plotted on a scale of .005 = 1. If we assume the velocity of entering equal to the velocity in the pipe $= c_1$, the inner radius of the wheel will be equal to that of the pipe and

$$2r_1 = \sqrt{\frac{4}{\pi} \times \frac{0.016}{2.0}} = .100 \text{ m.}$$

Hence in Fig. 3, $OI = r_1$ is laid off equal to .025 m. (the scale being $\frac{1}{2}$ full size), and $OII = r_2 = 2.5 \times r_1 = .0625$ m.

The absolute velocity of the entering water is represented by Ic_1; in like manner $IIR'' = c_{r_2}$ is the radial component of the velocity of efflux, which is equal to $c_1 \times \frac{b_1 r_1}{b_2 r_2}$, where b_1 and b_2 are the widths of the wheel at r_1 and r_2. In Fig. 3, c_{r_2} was taken $= c_1$, as is usual, which makes $IIR'' = Ic_1$.

On IIA assume v_2 of any convenient magnitude. It is then required to determine the vane corresponding with this assumed velocity. With A as center and AII as radius, describe an arc of a circle intersecting at point K_a another circle, struck with II as center and a radius equal to $\sqrt{2gh_a}$ (plotted on the scale of velocities).

$$\sqrt{2gh_a} = \sqrt{2g\zeta h} = \sqrt{2 \times 9.81 \times 1.5 \times 10.0} = 17.143 \text{ m.}$$

Project K_a up to A_1, when IIA_1 will give magnitude and direction of the absolute velocity of discharge c_2. For $\overline{IIK_a}^2 = IIC \times 2IIA$; hence $IIC = 2gh_a : 2v_2 = c_{r_2}$; therefore, since $CA_1 = IIR'' = c_{r_2}$, IIA' must be $= c_2$.

v_2 and c_2 being known, the parallelogram of velocities of discharge, $IIAA'A''$, may be drawn. The radial line Ao intersects Ia at a point a making $Ia = v_1$, and the parallelogram of velocities of entering water may be drawn $= IAc_1a''$. We have, therefore, determined the angles $\alpha = a''IT_1$ and $\beta = A''IIT_2$ which the vane makes with the inner and outer circumferences of the wheel. The vane itself is the arc of a circle satisfying these conditions.

For a second vane the velocity v_2 is taken such (= IIR), that

the circle struck from R with radius IIR intersects the circle struck from II (radius as before $\sqrt{2gh_n}$) in K_r so that RK_r is perpendicular to IIR: angle $RIIK_r$ was made $= 45°$. We then have $IIR' = c_2$, $IIR'' = w_2$, being radial; $\beta = 90°$. The parallelogram of entering velocities, Irc_1r'', was drawn as before, Ir being $= v_1$. The vane is shown by IR.

A further reduction of the velocity of rotation moves the point of intersection K higher up, until it reaches its highest position at K_h, when $v_2 = \frac{1}{2}IIK_h$. The parallelogram of discharge velocities is given by $IIHH'H''$ and $IIH = v_2$; $IIH' = c_2$, and $IIH'' = w_2$. As before we find $Ih = v_1$, and the parallelogram at point of entry is Ihc_1h'' The angles of the vane are $h''IT_1 = \alpha_h$ and $H''IIT_2 = \beta_h$, and the vane is represented by IH.

We will now explain the construction of the straight vane Ie. This separates the vanes curved back from those curved forward. Let $IIEE'E''$ and Iec_1e'' be the parallelograms at discharge and entry. For the vane in question the sides $E''IIt_2$ and $e''It_1$ are tangents to the same circle xx drawn with O as center. The direct construction of this vane is complicated, but the following approximate method is sufficiently close: We assume e'' to be very close to r'' and also, ER being a short length, $ER = E'R' = \frac{1}{2}E''R''$, and proceed as follows: Take $Ie''E$ close by Ir'' and produce it beyond I to t_1. Draw circle xx tangent to $Ie''E$, and draw the straight line t_2IIE'' tangent to this circle. Then make $ER = \frac{1}{2}E''R''$. $II\dot{E}$ will be the velocity of rotation of the wheel approximately; the parallelograms of velocities may be drawn as before and the direction of IE corrected if thought necessary.

Starting the centrifugal pump.—We shall now consider the starting of a pump filled with water to the height h. (Casing and pipe all considered as pump.)

To this end the velocity of rotation must be increased to a point v_g, at which the centrifugal action is sufficient to cause a fall of pressure at the inside circumference of the wheel, in consequence of which the atmospheric pressure on the lower level of water is just able to support the column, motion ensuing if the velocity is increased. Before discharge takes place we have,

therefore, equilibrium between the weight of the column of water, the hydraulic pressure at I and the pressure of the atmosphere; v_g will, therefore, be called the velocity of equilibrium. To maintain this equilibrium certain relations must exist between the pressures at the inner and outer circumferences of the wheel, which pressures are assumed due to heads h_{p1} and h_{p2} respectively, the heights h_u and h_o from the suction level to the wheel center and thence to the delivery orifice, the velocities of the wheel, v_{1g} and v_{2g}, and the atmospheric pressure h_b. These relations are expressed by the following equations:

$$\left.\begin{aligned} h_b &= h_u + h_{p1} \\ h_{p2} - h_{p1} &= \frac{1}{2g}(v^2_{2g} - v^2_{1g}) \\ h_{p2} &= h_o + h_b \end{aligned}\right\} \text{ whence } \frac{v^2_{2g} - v^2_{1g}}{2g} = h_u + h_o = h.$$

As $v_{2g} : v_{1g} = r_2 : r_1$, we get

$$v_{2g} = \sqrt{2gh\,\frac{r_2^2}{r_2^2 - r_1^2}} \qquad (4)$$

In Fig. 3 this velocity v_{2g} is determined graphically. Draw any radial line OaA; make $AB = aI$. Draw BL perpendicular to IIA, intersecting a circle drawn with A as center and AII as radius at L. Draw LM horizontal and ng parallel to MA, N being found from $IIN = \sqrt{2gh} = \sqrt{2 \times 9.81 \times 10.0} = 14.006$ m. IIG will then be the velocity of equilibrium v_{2g}. In proof we have

$$IIG : IIN = IIA : IIM.$$

Hence $IIG : \sqrt{2gh} = v_2 : BL.$

$$BL = \sqrt{BH(2AH - BH)} = \sqrt{(v_2 - v_1)(v_2 + v_1)}$$

$$\therefore\ IIH = \sqrt{2gh\,\frac{v_2^2}{v_2^2 - v_1^2}} = \sqrt{2gh\,\frac{r_2^2}{r_2^2 - r_1^2}} = v_{2g}.$$

For this velocity of wheel the parallelograms $IIGG'G''$ and Igc_1g'' were drawn and the corresponding vane IG.

All vanes following IG require for their normal running a velocity of rotation $v_2 > v_{2g}$; all vanes preceding IG require a velocity of wheel $v_2 < v_{2g}$. From this it is evident that for vanes belonging to the second group the velocity of starting must ex-

ceed that at ordinary working. This may cause inconvenience where the pump is worked off a shaft turning at a fixed uniform speed, the transmission to the pump being unchangeable. In such a case the flow must be started by lowering the initial head so as to make the normal velocity of rotation the starting velocity. Then

$$v_2 = \sqrt{2gh\,\frac{r_2^2}{r_2^2 - r_1^2}}$$

$$\therefore\ v_{2g} : v_2 = \sqrt{h} : \sqrt{h_o} \text{ and } h_o = h\left(\frac{v_2}{v_{2g}}\right)^2.$$

The inconvenience in starting due to $v_{2g} > v_2$ becomes more aggravated the greater the difference between v_{2g} and v_2, *i. e.*, the further the vane is removed from IG and the nearer it comes to IH.

Pressure relations.—Representing by h_{p1} and h_{p2} the head due to the hydraulic pressures at the inner and outer edges of the wheel, and the hydraulic resistances of the water as it traverses the wheel by z_r we have:

$$h_{p1} + \frac{w_1^2}{2g} + \frac{v_2^2 - v_1^2}{2g} = h_{p2} + \frac{w_2^2}{2g} + z_r.$$

Since $w_1^2 = c_1^2 + v_1^2$; $w_2^2 = c_{r2}^2 + (v_2 - c_{t2})^2$; and $c_{r2} = c_1$,

$$2g(h_{p2} - h_{p1} + z_r) = v_2^2 - (v_2 - c_{t2})^2 = c_{t2}(2v_2 - c_{t2}) \qquad (5)$$

In Fig. 3 for $v_2 =$ any convenient quantity,

$$= IIA,\ CK_a = \sqrt{IIC(2IIA - IIc)} = \sqrt{c_{t2}(2v_2 - c_{t2})}$$
$$= \sqrt{2g(h_{p2} - h_{p1} + zr)}.$$

From this it is apparent that the difference of head $(h_{p2} - (h_{p1} - z_r))$ decreases with v_2, becoming $= 0$ for a wheel with vane IH, because in this case the line corresponding with CK_a disappears. For this reason we have for vane H, $h_{p2} = h_{p1} - z_r$, less than h_{p1}.

If z_u is the head due to the resistances from the lower level to the inner circumference of the wheel, then

$$h_b = h_u + h_{p1} + \frac{c_1^2}{2g} + z_u \quad \therefore\ h_{p1} = h_b - \left(h_u + z_u + \frac{c_1^2}{2g}\right).$$

During ordinary running, for a wheel with vane IH, we shall have

$$h_{p2} = h_b - \left(h_u + z_u + \frac{c_1^2}{2g} + z_r \right).$$

As long as h_{p2} has a positive value, even if small, there is no difficulty. But when h_{p2} becomes $\gtrless 0$, the vane giving such a result is not a proper one. In such a case the water as it leaves the wheel separates bodily from the succeeding layers, is overtaken by them, and constant separation and shock is the result. As with every pump changes in velocity must be expected, we must select such a form of vane as will preclude the possibility of h_{p2} becoming $\gtrless 0$.

It is therefore interesting to determine the vane for which h_{p2} will $= 0$. From (5) we have

$$2g(h_{p1} - z_r) = c_{t2}(c_{t2} - 2v_2)$$
$$= 2g\left[h_b - \left(h_u + z_u + \frac{c_1^2}{2g} + z_r \right) \right] \qquad (6)$$

Assume $h_u = 4.0$ m.; estimate $z_u = 1.2$ m.;

$z_r = 1.2$ m.; $h_b = 10$ m., and $\frac{c_1^2}{2g} = \frac{2^2}{2g} = .20$ m.

Then for the wheel with vane IH

$$h_{p1} = h_b - \left(h_u + z_u + \frac{c_1^2}{2g} \right) = 10 - (4 + 1.2 + .2) = 4.6 \text{ m. and}$$
$$h_{p2} = h_{p1} - z_r = 4.6 - 1.2 = 3.4 \text{ m.}$$

For a greater value of h_u these heads would be still further diminished and for $h_u = 7.4$ m., h_{p2} would be $= 0$.

For $h_u = 4.0$ m. we have

$$\sqrt{2g(h_{p1} - z_r)} = \sqrt{2 \times 9.81 \times 3.4} = 8.168 \text{ m.}$$

Lay off this velocity on its proper scale on UU' (Fig. 3); with II as center describe the arc $U'T'$; make tangent $TT' =$ tangent UU' and draw $T'\mathfrak{T}$ perpendicular to IIT'. The velocity for the vane which marks the limit is given by $IIZ = \frac{1}{2}\, II\mathfrak{T}$, and the tangential component of the absolute velocity of discharge, c_{t2}, by IIT'. In proof we have

$$II T = \sqrt{2g\zeta h} = \sqrt{II\mathfrak{T} \times IIT'}$$

and $T'T = \sqrt{2g(h_{p1} - z_r} = \sqrt{T'II \times T'\mathfrak{T}}$

for $II\mathfrak{T} = 2v_2$ and $IIT' = c_{u2}$ (see (3) and (6)).

The parallelograms $II ZZ'Z''$ and Izc_1z'' and the vane IZ are drawn as before.

On account of uncertainty as to the correctness of the values assigned to z_u and z_r, the form of vane decided upon must not resemble IZ too closely. For this reason form IH is an undesirable one. IZ will the more closely resemble IH, the nearer T approaches K_h, *i. e.*, the smaller $TT' = UU' = \sqrt{2g\,(h_{p1} - z_r)}$, or the greater h_u, the height of the inlet above the lower level, or the greater $(z_u + z_r)$ or c_1. As shown above, IH coincides with IZ for $h_u = 7.4$ m. For a greater value of h_u we have $h_{p1} < z_r$, showing that the limiting vane (for which $h_{p2} = 0$) already falls between H and R.

For $h_u = 8$ m. h_{p1} will $= 0.6$ m., and according to (5)

$$\sqrt{2g(h_{p2} - h_{p1} + z_r)} = \sqrt{2g(0 - 0.6 + 1.2)}$$
$$= \sqrt{2 \times 9.81 \times 0.6} = \sqrt{11.77} = 3.43 \text{ m.}$$

If, therefore, K is taken on the circle $\sqrt{2\,gh_n}$ a distance 17.15 mm. $(= .005 + 3.43$ m.) below IIT' we should obtain v_2 equivalent to the radius of a circle passing through K and II with center on the line IIT'. This is greater than v_{2h}, the velocity of rotation corresponding to vane H. For this height on the suction side of the pump vane H would, therefore, be inadmissible, as the limiting form of vane, for which $h_{p2} = 0$ falls between H and R.

Hydraulic resistances.—The total head equivalent to the hydraulic resistances, z, may be regarded as made up of z_u, z_r, z_a and z_o, being the heads due to resistance in the suction pipe, in the wheel, in the conduit, and in the delivery pipe. z_u and z_o remain constant while the velocity of the water $c_u = c_o = c_1$, remains constant. z_r and z_a, however, vary for different forms of vanes. They may be represented approximately by the expressions $\zeta_r \frac{w_1^2 + w_2^2}{4g}$ and $\zeta_a \frac{c_2^2 + c_o^2}{4g}$ respectively. The change of

w_1 is of little weight compared with that of w_2 and c_2; and c_o is constant: for which reason the hydraulic resistances will be a minimum for the minimum value of $\zeta_r w_2^2 + \zeta_a c_2^2$; or, assuming $\zeta_r = \zeta_a$, for the minimum value of $(w_2^2 + c_2^2)$. But

$$(w_2^2 + c_2^2) = (c_{r2}^2 + w_{t2}^2) + (c_{r2}^2 + c_{t2}^2)$$
$$= 2c_1^2(v_2 - c_{t2})^2 + c_{t2}^2 = (2c_1^2 - 2g\zeta h) + (v_2^2 + 2c_{t2}^2),$$

remembering that $c_{r2} = c_1$ and $v_2 c_{t2} = g\zeta h$.

ζ is, strictly speaking, also variable for varying values of z_r and z_a, but this variation may be neglected for values near the minimum. We shall, therefore, obtain the velocity of rotation involving the least internal resistance as the one which makes

$$(v_2^2 + 2c_{t2}^2) = v_2^2 + 2\left(\frac{g\zeta h}{v_2}\right)^2 \text{ a minimum.}$$

Differentiating for v_2 and placing $= 0$

$$2v_2 - 4\frac{(g\zeta h)^2}{v_2^3} = 0.$$

This gives

$$v_2 = \sqrt{v_2 \times g\zeta h} = \sqrt{1.414 \times 9.81 \times 1.50 \times 10.0} = 14.42 \text{ m.}$$

For the most important forms of vanes as studied above v_2 had the values:

for	G	E	R		H
$v_2 =$	15.28	12.50	12.13	and	8.58

Vane M therefore, the one involving the minimum of internal resistance, would be found to lie between G and E.

The expression $w_2^2 + c_2^2 = 2c_1^2 - 2g\zeta h + v_2^2 + 2\left(\frac{g\zeta h}{v^2}\right)$ will have the following values for the different vanes considered:

for	G	M	E	R	H
$w_2^2 + c_2^2 =$	132.5	129.8	146.35	155.3	375.2

Similarly $(w_1^2 + c_o^2)$ is found to be:

$$(w_1^2 + c_o^2) = 45.6,\ 41.3,\ 32.6,\ 31.6,\ 19.6.$$

And the sum

$$(w_1^2 + w_2^2) + (c_2^2 + c_o^2) = 178.1,\ 171.1,\ 179.5,\ 186.9,\ 394.8$$

will give figures proportional to the heads due to internal resistances $(z_r + z_a)$. Or, expressed in terms of the resistance of R as unity,

0.954, 0.915, 0.960, 1.000, 2.112.

In Fig. 3 ζ, was uniformly assumed as 1.5;

therefore $h_n = 1.5h = h + z + \frac{c_o^2}{2g}$. $h = 10$ m., and $c_o = 2.0$ m.

Therefore $z = 1.5h - h - \frac{c_o^2}{2g} = 15 - 10 - \frac{2^2}{2g} = 4.8$ m.

Now let us assume that this value of ζ is the correct one for vane R.

Then it will not be correct for the other vanes owing to the changed values of $(z_r + z_a)$. To calculate these values and that of ζ, it is estimated that head $z = 4.8$ is divided for a pump with vanes R so that $(z_o + z_u) = 1.6$ m. and $(z_r + z_a) = 3.2$ m. Multiplying the fractions representing relative resistances by 3.2, we shall have $(z_r + z_a) = 3.05, 2.93, 3.07, 3.20, 6.76$ m.

Adding to these heads the constant one $(z_o + z_u) = 1.6$, we have $z = 4.65, 4.53, 4.67, 4.80, 8.36$ m.

Coefficient ζ is obtained from

$$\zeta = \frac{1}{h}\left(h + z + \frac{c_o^2}{2g}\right) = 1.02 + \frac{z}{10}$$

$$\therefore \zeta = 1.485, 1.473, 1.487, 1.500, 1.856.$$

To be exact the calculations for the different vanes should now be repeated using the corrected value of ζ. This correction is slight for G, M, and E_1 but considerable for H.

The results of this calculation, based on a number of suppositions, cannot be expected to give actual values. The assumptions were: 1. that $\zeta_r = \zeta_a$; 2. that for a wheel with vanes R the head due to internal resistances is exactly 4.8 m.; and 3. that this head is distributed so that $(z_o + z_u) = 1.6$ and $(z_r + z_a) = 3.2$. The calculation is reliable enough to show, however, that for vanes varying from form G to R the value of ζ suffers little change, but increases considerably as H is approached.

H and Z were obtained in the Fig. for $\zeta = 1.5$; they should be reconstructed for $\zeta = 1.8$ to 2.0. This would throw H and Z further to the left in the drawing and increase their velocities of rotation.

It is apparent that a form of vane approaching H is not to be

recommended owing to the increased hydraulic resistances which its use involves.

Loss of water by leakage between the wheel and the casing.—As the wheel does not fit the casing closely, some water will leak back from the discharge conduit into the suction space. This loss is proportional to the space left between wheel and casing and to the difference in pressure ($h_{p2} - h_{p1}$) existing at the inner and outer circumference of the wheel. The smaller this difference the less will be the loss, and in this respect wheels with vanes like *H* have the advantage. A little beyond *H* the pressure difference disappears, and from there on to *Z* it is negative, so that a reverse current would set in from the suction to the delivery chamber. However, in well constructed pumps the loss due to this leakage should, even with vanes *G*, not exceed a few per cent.

Calculation of angles of vanes and influence of the ratio of radii $\frac{r_2}{r_1}$. From Fig. 3 are obtained at once the equations necessary for solving for α and β. From the triangle of velocities Ic_1a'' we get

$$\cot \alpha = \frac{v_1}{v_2} = \frac{v_2}{c_1} \times \frac{r_1}{r_2} \qquad (7)$$

and from triangle $IIR''A''$

$$\cot \beta = \frac{v_2 - c_{r2}}{c_1}.$$

As $v_2 c_{r2} = g\zeta h$ $\therefore$ $c_{r2} = g\zeta h : v_2$,

$$\text{and } \cot \beta = \frac{v_2}{v_1}\left(1 - \frac{g\zeta h}{v_2^2}\right) \qquad (8)$$

For determining α and β for vanes *G*, *R* and *H* we must remember:

$$\left.\begin{array}{ll} \text{For } G \;\; . \;\; . & v_2 = v_{2y} = \sqrt{2gh\,\dfrac{r_2^2}{r_2^2 - r_1^2}} \\ \text{For } R \;\; . \;\; . & \beta = 90^\circ \text{ and } v_2 = \sqrt{g\zeta h} \\ \text{For } H \;\; . \;\; . & 2v_2 = c_{r2} = \dfrac{g\zeta h}{v_2},\ \therefore\ v_2 = \sqrt{\dfrac{g\zeta h}{2}} \end{array}\right\} \qquad (9)$$

In order to calculate the angles for vane E we must remember that the perpendicular dropped from O on IE, Fig. 3,

$$= OI \cos \alpha \text{ and } OE \cos \beta$$

$$\therefore r_1 \cos \alpha = r_2 \cos \beta \qquad (10)$$

The elimination of α and β from equations (7), (8) and (10) gives a cubic equation for v_2 which is difficult of solution. Proceeding as in the construction of the vane, we arrive at an approximate value as follows:

For vane IR, $v_{2r} = \sqrt{g\zeta h}$ and $\cot \alpha = \dfrac{\sqrt{g\zeta h}}{c_1} \times \dfrac{r_1}{r_2}$

Now assume α_e a little smaller than α_r and from (10) we obtain

$$\cos \beta_e = \frac{r_1}{r_2} \cos \alpha_e$$

$$\text{and } v_{2y} = v_{2r} + \frac{1}{2} R'E'' = \sqrt{g\zeta h} + \frac{c_1}{2} \cot \beta_e,$$

from which we might obtain a second closer value of α_e.

In order to examine the influence of a varying ratio of $\left(\dfrac{r_2}{r_1}\right)$ on the form of vanes and the velocities required, vanes G, E, R, H and Z, were in Fig. 4 constructed for $\dfrac{r_2}{r_1} = 2.0$ and 3.0. They are represented on half scale. The values of v_2 obtained are:

	G	E	R	H	Z
$r_2 : r_1 = 2.0$	16.174	12.650	12.130	8.578	7.700
$r_2 : r_1 = 2.5$	15.283	12.500	12.130	8.578	7.700
$r_2 : r_1 = 3.0$	14.856	12.450	12.130	8.578	7.700

For all wheels $r_1 = .05$ m. and $r_2 = 1.00, .125$ and $.150$. From the above velocities the revolutions per minute are calculated from $U = \dfrac{60 \times v_2}{2\pi \times r_2}$ and given in the following table:

	G	*E*	*R*	*H*	*Z*
$r_2 : r_1 = 2.0$	1,542	1,206	1,157	818	734
$r_2 : r_1 = 2.5$	1,167	955	927	665	588
$r_2 : r_1 = 3.0$	946	792	772	546	490

CONCLUSION.

The properties found above to be characteristic of certain forms of vanes may be summarized as follows:

1. A centrifugal pump to lift a given quantity against a given head may under normal conditions of working (water entering without shock) be run at a reduced speed of rotation in proportion as the vanes depart from the ordinary form (with a curvature backward) and approach the plane form or are curved forward.

2. The velocity necessary for starting is identical with the working velocity only for vane *G*. For vanes beyond *G* in the direction of *A*, $v_{2g} < v_2$, so that the water will be readily lifted while the wheel is being speeded up to its ordinary working speed; for vanes beyond *G* in the direction of *H*, $v_{2g} > v_2$, so that the wheel must be run faster than ordinary speed at starting. The increase over the normal speed is greater the more nearly the form of vane *IH* is reached.

3. The pressure existing at the outer circumference of the wheel is less the more nearly the vane approaches form *Z*, for which it disappears. For a moderate lift at the suction side *Z* is found beyond *H*; it approaches *H* as this lift increases, and may even fall between *H* and *R*. As a wheel so constructed will not work without constant shock and interruption of flow, the form of vane should never approach *Z*.

4. As regards hydraulic resistance there is little difference between vanes *G* to *R*; but vanes similar to *H* involve considerable increase.

5. As to loss due to leakage through the clearance space

Fig. 3.

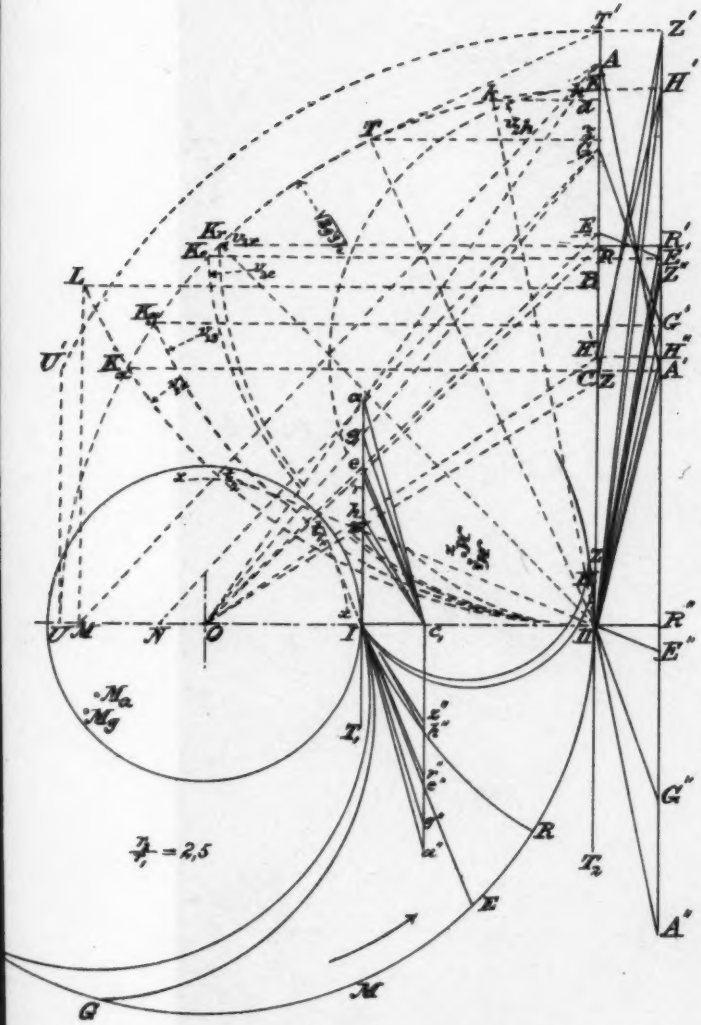


Fig. 1.

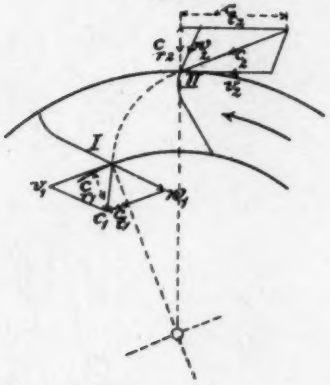


Fig. 2.

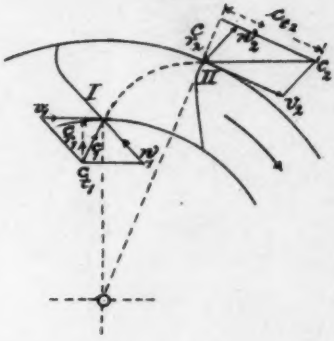
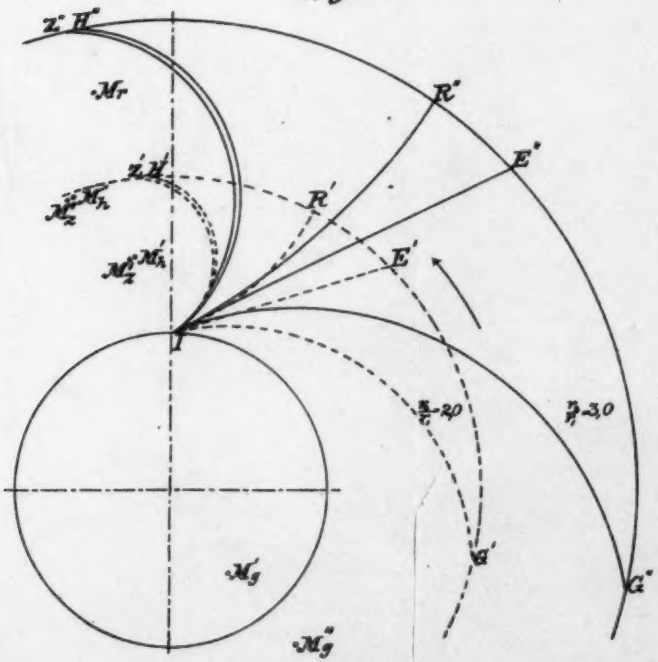


Fig. 4.



between wheel and casing, wheels with vanes like *IH* have the advantage. With well constructed wheels this loss is, however, trifling.

From what has been said it appears that vanes shaped like *H* have only the advantages of smaller rotative speed and loss by leakage. Beside the forms found most commonly in use, *G* and *R*, the plane vane *E* is to be recommended. It has the advantage of great simplicity, and should be used in preference to *G* and *R* wherever a slightly increased velocity at starting may be readily imparted.

XXXV.

THE RATIONAL LUBRICATION OF MARINE ENGINES.

Under this title an article appeared in the March (1891) number of the "Revue Maritime et Coloniale" by Chief Engineer G. Fontaine of the French Navy, from which the following is selected: In order to make a comparison between the quantity of material for internal lubrication consumed by different types of engines, the most accurate way to proceed would be to form for each type the product of the surface to be lubricated by the number of revolutions. These products would be proportional to the quantities required, any extraordinary features receiving their appropriate weight. Assuming, however, for different types, the same pressure per square inch of bearing surface, and a constant ratio between length and diameter of journals, we may take the lubricating material required proportional to the product of the square of the shaft diameter, times revolutions per minute, times number of cylinders. The two-cylinder engines of 1865 consumed about 1.6 kg. of lubricant per ton of coal burnt, including

.6 kg. of tallow. The best modern engines use about 3 kg. of oil, of which 0.1 kg. is for internal lubrication. Considering that about twice the power is now developed per ton of coal than was developed in 1865, and considering also the increase in complication of parts and of piston speed, we conclude that the amount of oil used for external lubrication per I.H.P. has remained about constant, while that for internal lubrication is now but one tenth of what it was formerly. The division of engine power brought about by use of twin screws increases the amount of lubricant necessary about 35 per cent. A further increase is due to the independent working of auxiliary machinery which, though it represents an aggregate power of not more than from $\frac{1}{30}$ to $\frac{1}{20}$ of the main engine power, yet requires double the oil of the latter per I.H.P., so that its consumption of lubricant is from $\frac{1}{25}$ to $\frac{1}{10}$ that of the main engines.

According to the amount of lubricant they require, engines may be classified as follows:

a. Engines with two vertical cylinders without independent auxiliaries. Consumption of oil 2 kg. per ton of coal.

b. Ditto, with independent auxiliaries. Consumption 2.2 kg. per ton of coal.

c. Engines with three vertical cylinders with independent auxiliaries. Consumption 3 to 3.3 kg.

d. Engines with three horizontal cylinders. Consumption 3 to 4 kg.

e. Engines with six vertical cylinders arranged tandem fashion, with independent auxiliaries. Consumption 5 to 6 kg. and over.

f. Small, fast running engines, like those of torpedo boats and cruisers, including auxiliary machinery for ventilating, lighting and operating the torpedo machinery, consume from 10 to 16 kg. per ton of coal, the smaller value referring to two-cylinder, the larger to three-cylinder engines.

Any oil consumption, irreconcilable with the particular type of engines and the manner in which they are used, whether on long trips, or on frequent short ones, at full power, or at reduced power, must be accounted for either by

1. Defective utilization for certain engine parts.

2. Defective distribution.

3. Waste brought about by considerations of safety.

4. Waste due to poor management, poor condition of the oiling apparatus, or of the rubbing surfaces, and to a number of other causes.

The first of these is the most important.

A journal enclosed in a bearing made in halves may serve as a type of a movable engine part. The clearance of the journal in its bearing is regulated to a nicety and kept filled with oil, whose office it is to interpose itself between the metallic surfaces. This object is attainable approximately when the pressure per sq. cm. does not exceed 28 kg. for a velocity of .3 m. per second, and 10 kg. for a velocity of 1.5 m. per second. The clearance allowed is a function of the diameter of the shaft, being equal to .002 of this diameter. The length of the bearing is an important factor; for an oil that will serve well in a short bearing may be forced out in a longer one, resulting in an abrading of the softer metal. The metallic particles abraded are mixed with the oil, which then becomes useless and should give place to a fresh supply.

The greatest shaft diameters in use are about .4 m.; the play in the bearings is therefore .8 mm.; required therefore an oil of such consistency and cohesive quality as to support itself in a space .4 mm. wide and .4 m. high. With such an oil the proper play may be allowed without danger of thumping. If an oil is used not viscous enough to fill this space to the top, the bearing will run warm and more will have to be supplied to keep its level high. The temperature of the engine room and the season of the year enter into this consideration, and bearings which may thump during the summer months, may, with the same quantity of the same oil run smoothly during the winter. For the large bearings of marine engines, particularly of ships trading in the tropics, mineral oil and, of vegetable oils, castor oil are best suited. Experiment has shown that 2 kg. of castor oil are equivalent to 3 of colza oil and to 2.6 of olive oil. For small bearings and for auxiliary machinery a more liquid oil, like colza or olive oil, may be used.

For the preservation of machinery not in use the oil should

have the qualities of adhesiveness and unchangeableness. Mineral oil and, best of all, vaseline are recommended.

The proper location of the oil holes has an important bearing on economy. It stands to reason that with a horizontal bearing the oil will flow out equally at the two ends. If, however, there is more than one oil hole in the length of the bearing, the oil will escape to the two sides, while the lubricant filling the clearance space between the oil holes cannot so readily find a way out. Abraded particles of metal gradually thicken it, until in the end the bearing will run warm, although the oil supply may be over abundant. This evil is overcome by leaving a space between the upper and lower halves of the bearing or by cutting a channel connecting the oil holes. Increasing the number of oil holes is not a good plan, because it may be regarded as a fixed principle, that to insure economy the oil should remain in contact over as great a length as possible. The oil holes should therefore be as close together as possible, and if a single one is used, it should be of ample bore and enlarged at the bottom. For horizontal bearings the oil hole should be in the middle of its length; for inclined bearings it should be near the upper end, and nearer the end the greater the inclination.

Occasionally, to prevent waste, when using oils too liquid, water is introduced by drops near the ends of the bearing. This forms an emulsion with the oil preventing its outflow until a temperature of from 95 to 105 degrees Fahr. is reached, when the two substances separate, allowing a freer flow of oil, and in this manner acting as a regulator. The economy resulting from this experiment is sometimes ascribed to the water, but without reason, for the coefficient of friction using water as lubricant is twice as great as that with oil. A form of oil cup facilitating the use of viscous oils by taking advantage of the change of temperature for automatic regulation is described. It consists of a cup containing the oil, into which fits a second cup, which may be more or less weighted by dropping shot into it, for regulating the feed. Should the bearing run warm, the oil, growing more fluid, will feed more rapidly. A cover attached to the shot receptacle, passing over the outside of the oil cup, protects the oil from dust.

The merits and demerits of lubrication by means of wicks are too well known to be repeated. The advantages of a sight feed system are: possibility of being kept in good order; oil supply securely stored; avoidance of waste due to use of portable oiling gear. The disadvantages are: irregularity of supply to the different engine parts; greater direct consumption than with wicks. The oil never being perfectly free from impurities and glutinous matter tends to choke the lengthy oil pipes, and the supply must be constantly regulated by manipulating the valves. Besides it is not an easy matter to gauge the proper number of drops for every part and for the varying engine speeds. All this entrains wastes inseparable from the system. To take advantage of the good points of both systems, wicks and sight feeds, the author recommends the use of wick cups in conjunction with the sight feeds. To facilitate the cleaning of the oiling system, the main distributing pipe should be fitted with a draincock, and a connection for blowing through with steam. For thrust bearings a heavy oil is used, and danger of heating met by either surrounding the bearing by a casing through which water is made to circulate, or by forming the bearing into a trough containing water to a certain height, which is covered by a layer of oil, and is constantly renewed.

Friction is not only dependent on the nature of the materials used; the condition of the surfaces is an important factor. Hammering the surfaces being machined before the last cut is taken is a good practice, as is also the superficial hardening of journals by subjecting them to the pressure of cylindrical rollers, or the case-hardening of iron parts. White metal is much improved by hammering, and some concerns fit their bearings with white metal by hammering into recesses rather than by pouring. This method avoids the necessity of taking mandrels to sea on ships. Excellent results have been obtained by hammering white metal plugs into the bearing faces of piston rings and valves.

Although oil grooves decrease the bearing surface, yet experience has demonstrated beyond a doubt that they greatly diminish the wear.

On putting a ship, that has been laid up for a time, on active

duty, it is important to discover by actual trial whether everything about the machinery is in condition to function properly under full power. Dock trials under full steam, where such can be made, are an even more severe test than a trial under way, for with equal I.H.P. developed the number of turns in the former case will be less than in the latter, the mean pressures will therefore be greater and the strains on the bearings more severe. If n is the number of revolutions per minute which the engine would make under way to develope a given I.H.P., and n_1 the revolutions for the same I.H.P. when making a dock trial, then

$$\frac{n^3}{n_1^3} = \frac{1.54}{1}.$$

II. INTERNAL LUBRICATION.

For internal lubrication heavy mineral oils are used. Only a small fraction of the oil fed, namely only that portion which, together with a film of water, is deposited on the working faces, is put to use. The rest passes through to the condenser. The water film aids very much in the work of lubrication, and some torpedo boat engines are not oiled at all. The oil allowed for internal lubrication in the French Navy is .005 kg. of mineral oil per square meter of surface to be lubricated per revolution, the highest limit being .25 kg. per ton of coal burnt. On merchant steamers the consumption of oleonaphtha varies between .085 to 1.2 kg. per ton of coal; of valvoline about one-half this amount is sufficient.

It is a good plan to leave shallow depressions in the wearing surfaces of valves, and for small valves these may be all that is required for balancing, as the pressure of the steam contained in them is sufficient to balance the pressure on the back. Piston rings should be soft and not too stiff. All cylinders wear larger in the middle, so that, as the piston travels from this point toward either end, the rings must be compressed, and if they are too strongly set out, the work of contraction is considerable.

It is the opinion of many that, after a ship has been at sea three days, the amount of oil for internal lubrication may be de-

creased ten per cent. each day until it has been reduced one-half.

Of oiling apparatus those that operate through condensation of steam are superior to oil pumps, in being more regular and in cleaning the oil somewhat as it passes through the water. If, on opening up the cylinder and valve chest, the surfaces feel greasy to the touch, showing bright under the film, the oil supply was excessive; a scanty oil supply is evidenced by a certain dryness of the surfaces. All polished surfaces should be found free from surface cracks, and all other surfaces free from rust. If, after the engine has been opened for some days, the surfaces of the high-pressure valve chest show signs of rust, while the cylinders are greasy, the oil used was too volatile and was not atomized until it reached the cylinder.

Too much oil used in the cylinder will be made apparent in the condenser by the appearance of the tubes. Those near the exhaust pipe nozzle will appear bright, as if polished, and on those next the nozzle will be found adhering drops of thickened oil. With a rational lubrication the tubes should show a uniform oily film. Excess of oil will also show in the hot well, and cause the rubber valves of the pumps to swell.

XXXVI.

THE SPONTANEOUS IGNITION OF COAL.

The following is an abstract of a paper read before the British Association at Cardiff by Professor Vivian B. Lewes:

It has been held for a long time that the pyrites (iron disulphide) contained in coal were the chief cause of spontaneous ignition, brought about by the heat engendered by the oxidation of the sulphide into sulphates of iron. Iron pyrites is found in coal in several different forms: sometimes as a dark powder distributed

throughout the mass of the coal, and scarcely to be distinguished from the coal itself. In larger quantities it is often found forming thin golden-looking layers in the cleavage of the coal, while it sometimes occurs as large masses and veins, often 1 to 2 inches in thickness. Such masses of this impurity are, however, mostly picked out at the mine. In dry air the pyrites undergo but little change at ordinary temperatures; but in moist air they rapidly oxidize when in a finely divided condition. The heat evolved is not, however, sufficient to cause ignition. Some coals contain only .8 per cent. of pyrites, and rarely more than 2 per cent. If all the pyrites contained in 100 pounds of coal were concentrated in one spot and rapidly oxidized, the temperature would barely be raised to 100° C., if all loss of heat could be avoided. Besides which iron pyrites may be, and in some manufactures are, kept in a state approaching powder in heaps containing many hundred tons; and although undergoing oxidation the author never discovered a case of heating, much less a rise of temperature which would approach the igniting point of coal. When, however, it is mixed with finely divided coal, then heating and ignition is a frequent occurrence in even moderate-sized heaps.

Temperatures of ignition of different kinds of coal:

Cannel coal,	698° F.
Hartlepool coal,	766° F.
Lignite,	840° F.
Welsh steam coal,	870.5° F.

and the oxidation of the iron pyrites is totally unable to bring about increase of temperature to reach these.

Carbon possesses in a high degree the power of attracting and condensing gases upon its surface. The volume of oxygen that different coals are capable of absorbing varies from 1 ¼ to 3 times the volume of the coal. Charcoal has this property in so extraordinary a degree that even small quanties of it, newly made and powdered, will often ignite spontaneously. Heaps of 100 bushels or more will always ignite. Air-dry coal absorbs oxygen more rapidly than wet coal. It has been suggested that very bituminous coal, such as cannel coal, shale, and coal containing schist, is liable to spontaneous ignition from the fact that a rise in tem-

perature would cause heavy oils to exude from them, which by undergoing oxidation, might cause rapid heating. But experiment not only shows that this is not the case, but that the heavy mineral oils have a remarkable influence in retarding heating, cotton waste, oily with easily oxidizable oils mixed with 20 per cent. of heavy mineral oil, being exempt from heating. The oxygen absorbed by the coal is in a chemically active state, and less diluted with nitrogen than it is in the air, because nitrogen is not absorbed to the same extent. The absorption is always attended with evolution of heat, which in turn hastens the process, and this heat, in conjunction with the other conditions, all favorable to chemical action, bring about an oxidation of the hydrocarbons of the coal, generating more heat. Under ordinary conditions the heat so generated, being produced slowly, is dissipated as fast as produced; but in the center of a mass of coal, which with the air spaces forms an admirable non-conductor, the temperature may rise to the igniting point of coal.

The most important point to be noticed is the extraordinary effect which initial temperature has on the rapidity of chemical actions of this kind. At a low temperature and up to 100° F. the action will go on slowly, with practically little or no chance of undue heating taking place, but directly the temperature exceeds 100° F., then with some classes of coal ignition is only a question of time and mass. Though the igniting point of various coals lies above 700° F., yet if many of these coals are powdered, and are placed in perforated zinc cases in masses of 2 pounds or upwards, and these are kept at a steady temperature of 250° F., in an oven, ignition will generally follow in a few hours, whilst between this and 150° F., it will take days instead of hours for the same result to follow, and at ordinary English temperatures the mass of broken coal must be very great before ignition will take place.

In the case of coal cargoes the liability to spontaneons ignition increases with:

1. The increase in bulk of cargoes. In cargoes to places beyond Europe the cases of ignition amount to ¼ per cent. for cargoes under 500 tons; to 1 per cent. in cargoes from 500 to

1,000 tons; 3.5 per cent. for 1,000 to 1,500; 4.5 per cent. for 1,500 to 2,000; 9 per cent. for cargoes of 2,000 tons or over. Mass influences this action in two ways: (*a*) The larger the cargo the more non-conducting material will surround the spot where the heating is taking place. (*b*) The larger the cargo the greater will be the breaking down action of coal coming down the shoot upon that already loaded, and the greater therefore the absorbing surface.

2. The ports to which shipments are made. 26,631 shipments to European ports resulted in only 10 casualties; 4,485 shipments to Asia, Africa, and America resulted in 60. This great diversity is due partly to greater length of time the cargo is under shipment in the latter cases, and in great measure to the heat of the tropics hastening the chemical action.

3. The kind of coal which the cargo consists of.

4. The size of the coal; finely divided coal being much more likely to ignite than lump coal.

5. Shipping coal rich in pyrites when wet. The effect of the moisture is to retard the absorbtion of oxygen at first. It favors the oxidation of the pyrites, however, which swelling split the coal and increase the absorbing surface. It is in this way that the presence of pyrites tends to increase the list of spontaneous ignitions.

6. Ventilation of the cargo. For ventilation to do any good, cold air would have to sweep continuously and freely through every part of the cargo—a condition impossible to obtain in coal cargoes—while anything short of that only increases the danger, ordinary methods of ventilation supplying just about the right amount of air to create the maximum amount of heating. The reason is clear. A steam coal absorbs twice its own volume of air, and takes about ten days to do it under favorable circumstances. Retarding this process by excluding air is therefore a better plan than supplying by ventilation enough to saturate the coal and not enough to carry off the heat produced.

7. Rise in temperature in steam colliers due to the introduction of triple expansion engines and high pressure boilers. The

increase of temperature in the fireroom due to this is from 5 to 10 degrees, and this affects the adjacent parts.

In the case of coal bunkers in modern steamers and warships the mass factor may be dropped from the causes operating to bring about ignition, as the amount of coal in one bunker may be taken to be from 50 to 400 tons, and it has been shown that the cases of spontaneous ignition in masses of coal less than 500 tons does not amount to more than ¼ per cent., and the question of initial temperature becomes the most important factor. That this is the case is shown by the fact, based on observation, that bunker fires are almost entirely confined to vessels in which the bunker bulkheads are only separated from the smokepipe by a narrow air-space, or in close proximity to the boilers themselves. Taking the case of a fast liner, it is found that the temperature in a coal bunker varies very considerably according to its proximity to the air channel around the uptakes. Close to the outside of the bulkhead the temperature is often as high as 200°, whilst inside 120 would be a fair estimate. From the center of the bunker to the side the temperature is seldom above 75°, while near the iron deck it may rise to 100° owing to heat conduction. It has been pointed out that if coal is kept at a high temperature, even though it be far below the igniting point, ignition is only a question of time, and if the bunker coal next the bulkhead is kept at 120° F., any coal with a tendency to absorb oxygen will run a great chance of igniting within a few days. It is manifest that if this is the real cause of ignition, the seat of fire ought to be found near the bunker bulkhead, but this is not always the case, the mass of fire being often found near the center of the bunker or even well out to the ship's side. In these cases, however, a line of charred coal is usually found running from the heated bulkhead to the seat of the fire, showing that the fire, started by the high initial temperature, had not sufficient air near the bulkhead to more than smoulder, but as soon as it came in contact with a current of air, passing up through the coal from the hatches in the decks, the surrounding mass began to burn fiercely.

The tendency to spontaneous ignition in coal has always been supposed to increase with the quantity of pyrites in it, but experi-

ment shows that it is the tendency or power of the coal to absorb oxygen which must be taken as the true index of danger, and this may be roughly gauged by the amount of moisture which the coal has absorbed from the air, as, if much moisture be found in an air-dried sample of coal, it at once stamps it as a highly absorbent form and which must on that account be stored with special precaution, whilst if but little moisture be present, it is probably unable to take up enough oxygen to lead to serious mischief. This is shown by the following table, which also shows how little the pyrites have to do with ignition:

Liability to spontaneous combustion.	Pyrites. Per cent.	Moisture. Per cent.
Very slight	1.13	2.54
	1.01 to 3.04	2.75
	1.51	3.90
Medium	1.20	4.50
	1.08	4.55
	1.15	4.75
Great	1.12	4.85
	0.83	5.30
	0.84	5.52
	1.00	9.01

When once coal has taken up oxygen, and the early stages of heating are passed and the temperature has again fallen, all danger of ignition is over and it may be stored in any quantity with perfect safety, so that if it were practicable to keep newly won coal for a month in moderate sized heaps, and then to avoid much breakage in afterwards loading it, spontaneous ignition would be almost unknown.

The following Table 3, Speed Trials, completes the series commenced in the August issue of the JOURNAL:

	Name of Vessel.	*Philadelphia.*	*San Francisco.*	*Char*
Hull.	Duration of trial	Four hours.	Four hours.	Four
	Mean displacement of hull: tons	4,325	4,088	3,55
	Length between perpendiculars	315	300	30
	Beam at L.W.L.	48–7	49–1½	46
	Ratio of length to breadth	6.49 to 1	6.308 to 1	6.49
	Mean draught	19–2½	18–9	17–
	Area of L.W.L. plane	10,875	10,467	9,70
	Area of immersed midship section	815	770	74
	Coefficient of fineness of displacement	.515	.509	
	Wetted surface—Kirk's analysis	20,100	19,006	17,80
	I.H.P. per 100 square feet of wetted surface*: knots	43.85 19.67½	50.41 19.52½	37.43
	$\frac{S^2 \times \text{immersed midship section}}{\text{I.H.P.}}$: knots	716.67 19.67	598.72 19.52	666.01
	$\frac{S^2 \times D^{\frac{2}{3}}}{\text{I.H.P.}}$: knots	232.89 19.67	198.63 19.52	217.97
Steam.	Steam at boilers, per gauge	159.8	135.3	
	Steam at engines, per gauge: S.: P.	151.2 154.0	126.2 128.4	85.9
	Steam at first receiver, absolute: S.: P.	57.61 56.5	55.1 58.1	23.84
	Steam at second receiver, absolute: S.: P.	24.52 24.3	19.25 18.25	...
Engines.	Revolutions of main engines per minute: S.: P.	119.58 119.55	125.79 123.83	115.35
	Piston speed in feet per minute: S.: P.	797.2 797.0	754.78 743.65	692.10
	Position of throttle: S.: P.	Wide. Wide.	.55 .55	Wide.
	Cut-off in main cylinders, starboard: H.P: I.P: L.P.	.6 .72 .74	.69 .69 .69	.72
	Cut-off in main cylinders, port: H.P: I.P: L.P.	.58 .74 .75	.69 .69 .70	.72
	Mean pressure in main cylinders, starboard: H.P: I.P: L.P.	61.95 21.3 8.91	51 37 25.3 9.93	42.20
	Mean pressure in main cylinders, port: H.P: I.P: L.P.	57.73 19.9 9.98	46.45 26.8 10.64	44.75
	Aggregate equivalent mean pressure on L.P. piston: S.: P.	30.51 30.1	30.32 30.68	26.54
Vacuum.	Vacuum in condenser in inches: S.: P.	24.3 24.67	25.7 26.1	26.17
	Cooling surface per I.H.P.*	1.540	1.494	
Air pump.	Displacement of air pump piston per minute per I.H.P.*	.238	.305	
	Revolutions of air pump per minute: S.: P.	105.3 118.2	197. 131.	134.7
	I.H.P. of air pumps: S.: P.	31.56 33.32	66.48 41.23	21.98
Circulating pump	Revolutions of circulating pumps: S.: P.	274.8 234.3	199.7 201.25	343.3
	I.H.P. of circulating pumps: S.: P.	30.27 24.7	14.36 15.07	51.75
Blowers.	Revolutions of blowers per minute	516.	478.	48
	Number and aggregate I.H.P. of blowers	Eight: 87.57	Eight: 110.67	Four:
	Diameter of blower fans	4–6	3–4	3–4
	Air pressure in fire rooms in inches of water	1.49	2.02††	
I.H.P.	I.H.P. of H.P. cylinder: S.: P.	1,645.10 1,560.47	1,590.06 1,427.92	1,426.61
	I.H.P. of I.P. cylinder: S.: P.	1,352.17 1,325.24	1,603.63 1,700.43	...
	I.H.P. of L.P. cylinder: S.: P.	1,249.73 1,400.46	1,594.39 1,664.29	1,732.92
	I.H.P. of each main engine: S.: P.	4,247.0 4,286.17	4,788.08 4,792.64	3,159.53
	I.H.P. of main engines air and circulating pumps	8,653.02	9,717.86	6,45
	I.H.P. of feed pumps	34.48	42.85	4
	I.H.P. of all other auxiliaries	127.26	132.22	10
	Aggregate I.H.P. of all machinery	8,814.76	9,912.93	6,66
	Cubic feet per minute swept by L.P. piston per I.H.P.¶	7.54 7.47	7.59 7.47	8.64
Temperatures.	Temperature of fire room: forward: aft	117.6	114.1	9
	Temperature of engine room: forward: aft	87.6	100.7 94.8	107.5
	Temperature of injection	63.3	63.7	6
	Temperature of discharge: S.: P.	113.3 111.4	101.2 99.3	107.4
	Temperature of feed: S.: P.	145.4 134.3	109.5 103.1	116.4
Thrust.	Indicated thrust in pounds: S.: P.	57,481. 58,017.	66,988.5 68,115.4	51,651.2
Propeller.	Indicated thrust per square inch of thrust surface	47.66 48.11	59.02 60.01	53.9
	Indicated thrust per square foot of hel. area of propeller	1,006.7 1,016.1	1,163.19 1,182.76	943.4
	Constant for propeller area—K●	8.66	9.14	10
	Constant for propeller diameter—K¹: ‡	26,852.	21,774	21,875
	Slip of propeller, per cent.: S.: P.	18.2 18.2	15.45	8
Coal.	Coal used per hour: pounds	26,354.4	...	20,096
	Coal used per hour per square foot of grate	42.2	...	46
	Coal used per hour per square foot of heating surface	1.288	...	1
	Coal used per hour per I.H.P.: [all machinery]	2.99	...	3
Ratios.	Total heating surface per I.H.P.●	2.321‖	2.03	2
	I.H.P. per square foot of grate surface	14.126	17.46	15
	I.H.P. per ton of weight in engine compartment‡	23.63	22.07	19
	I.H.P. per ton of weight in boiler compartment‡	17.555	21.32	16
	I.H.P. per ton of weight, propelling machinery and water	10.35	11.25	9
	I.H.P per ton of weight, all machinery, spare gear and tools	10.08	10.84	9
	Ratio of cooling surface to total heating surface	1 to 1.514	1 to 1.386	1 to 1.
	Capacity of steam room in revolutions: in seconds†	38.6 19.3	33.77 16.2	33.71

● $K = \frac{\text{Hel. area}}{\sqrt{\frac{\text{I.H.P.}}{\text{Revolutions}}}}$ ‡ $K^1 = \frac{\text{Diameter}}{\sqrt{\frac{\text{I.HP.}}{(\text{Pitch} \times \text{Revs.})^3}}}$ † At given cut-off. ‡ Including water. ¶¶ Air, circulating,

TABLE 3.—SPEED TRIALS.

arleston.	*Baltimore.*	*Newark.*	*Yorktown.*	*Concord.*	*Dolphin.*	*P*
ur hours.	Four hours.	Four hours.	Four hours.	Four hours.	Six hours.	Fou
,557	4,500	3,980	1,703	1,740	1,413	
300	315	310 10	228	228	240	
46-2	48-6	49-2	36-0	36-0	32	
.49 to 1	6.495 to 1	6.305 to 1.	6.33 to 1	6.33 to 1.	7.5	5.6
17-10½	19-10½	18-3	14-0	14-1½	13-9½	
,700	10,875	10,500	5,765	5,765	4,717	3,8
741	816.64	775	435.3	439.8	338	
.51	.515	.510	.497	.502	.5015	
,806.6	20,385	19,110	11,428	11,510	10,250	6,7
18.2‡‡	48.89 20.096‡‡	46.41 19.0‡‡	32.31 16.14‡‡	29.6 15.8§	21.98 15.5§	15.4
18.2	695.3 20.096	613.06 19.0	539.6 16.14	509.8 15.8	558.63 15.5	460.18
18.2	261.1 20.096	198.62 19.0	176.75 16.14	167.4 15.8	208.03 15.5	153.28
91.44	Gauges out of order.	161.8	150.5	164.3	87.08	
86.3	125.1 123.44	156.1 152.6	142.3 144.8	159.9 159.6	85.21	
23.52	62.2 59.28	57.4 59.7	68.7 72.7	79.5 77.0	28.04	
...	20.4 21 75	16.8 16.6	20 6 19.7	29. 28 5	...	...
213.95	116.42 116.08	127.4 126.7	157.98 153.94	151.5 153.3	74.15	12
683.7	814.94 812.56	849. 844.4	789.9 779.7	157.3 766.7	293.2	69
Wide.	Wide. 7.5 holes.	.8 .8	Wide. Wide.	Wide. Wide.	Wide.	Wi
.67	.69 .77 .77	.75 .76 .62	.76 .76 .76	.79 .78 .78	.536 .602	.65
.67	.70 .77 .77	.72 .72 .65	.76 .76 .76	.79 .78 .78	..	...
14.55	50.17 21.87 9.52	58.8 25. 15.0	57.67 32.8 10.02	53.8 30.4 15.36	41.41 12.95	51.3
14.96	52.29 22.49 9 84	53.0 24. 13.6	55.65 35.5 10.72	53.1 28.7 15.0	...	...
26.83	28.23 29.20	38.23 35.28	33.61 34.94	37.26 36.1	24.96	28
26.06	24.5 24.3	25.9 25.6	24.84 25.02	24.1 24.	25.5	23
2.085	1.256	1.411	1 403	1.446	1.775	2
.405	.677	.244	.229	.214	.199	
135.2	116.42 116.08	105. 125.	123. 120.5	102.8 101.8	30.71	126
20.87	... ...	29.9 31.6	60.88¶¶ 57.91¶¶	13.73¶¶ 13.59¶¶	41.2¶¶	...
313.9	177 177	208. 209.	123. 120.5	102 8 101.8	30 71	282
41.56	23.21 23.21	15.06 15.14	... ...	... ...	††	4
486.	566.	663.	751. 763.	497.6	...	660
75.57	Eight: 110.86	Eight: 117.32	Four: 52.03	Four: 34.7	Four: 60. ½	Two:
3-4	4-6	4-6	4-6	4-6	6-0	4-6
1.77	2.085	2.23	1.25	2.29**	Fire rooms open.	4
1,395.45	1,682.65 1,749.17	1,353.5 1,223.0	519.76 495.41	464.7 464.0	1,021.95	504
...	1,514.10 1,555.69	1,355.2 1,310.6	588.78 631.89	524 7 502.0	...	...
1,761.08	1,625.62 1,680.61	1,739.6 1,574.3	471.16 498.28	688.7 670.10	1,122.22	496
3,156.53	4,845:59 4,985.48	4,468.3 4,113.9	1,579.7 1,625.58	1,678 1 1,636.10	2,144.17	1,000
,452.22	9,877.49	8,674.07	3,324.37	3,341 6	2,185.37	1,004
45.84	28.81	53.	16.15	17.6	7.75	3
168.10	181.34	141.5	52.03	45.3	60.0	36
,666.16	10,064.42	8,868.5	3,392.55	3,404.5	2,253.12	1,044
8.54	8.45 8.19	6.01 6.48	6.79 6.52	6.13 6.29	8.73	7
92.3	93. 93.	66.3	58.7 72.5	91.4	87.3 97.1	96
90.1	70. 73.	86.1 70.3	71. 84.2	89.3	85.86	93
65.3	56.	45.	40.4	36.	67.	77
106.4	100. 97.	103.3 73.3	97.2 88.0	105.	96.3	116
115.1	129. 131.	112.4 78.3	91. 102.6	139.9	111.19	119
52,236.2	63,890. 65,920.	61,026. 56,489.	26,398 27,521.	27,692. 26,670.	39,756.	21,142
54.5	61.2 63.15	47.98 44.41	72.88 75.98	52.06 50.1	43.72	55
954 08	1,317.6 1,153.0	1,156.3 1,070.4	1,837.5 1,083.0	1,044 1,005 6	551.1	898
10 16	8.69	8.93	7.70	8.03	13.09	8.
,875.	18,012.	26,071.	22,340.	23,000.	22,537.	18,750.
8.	18.62 18.38	20 3 19.87	17.19 16.07	19.9 20.8	11.65	24
,096.88	30,353.88	21,590.5	...	9,402.1	4,986.	4,275
46.07	44.9	40.0	...	42.7	18.25	42.
1.290	1.767	1.290	...	1.145	.752	1.
3.01	3.15	2.43	...	2.76	2.19	4.
2.33	1.707	1.887	2 385	2.4117	2.91	2.
15.28	14.89	16.42	15.42	15.47	8 34	19.
19.76	21.61	25.48	24 10	24.9	11.758	17.
16.88	20 51	21.36	16.72	15.47	11.71	15.
9.72	10.53	12.08	10.46	9.92	5.867	8.
9.15	10.12	11.62	10.23	9.59	...	7.
to 1.129	1 to 1.360	1 to 1.337	1 to 1.7	1 to 1.931	1 to 1.64	1 to 1.
17.6	33.98 17.6	32.6 15.45	59.8 23.0	63.02 24.8	33.5 27.13	24 5

ng, and bilge pump. | Auxiliary boiler not used. • Aggregate I.H.P. §§ Estimated. •• Closed ash-pits. ¶ Main engines. ‡ Meas

TABLE 3.—SPEED TRIALS.

San Francisco.	*Charleston.*	*Baltimore.*	*Newark.*	*Yorktown.*	*Concord.*
Four hours.	Four hours.	Four hours.	Four hours.	Four hours.	Four hours.
4,088	3,537	4,500	3,980	1,703	1,740
300	300	315	310 10	228	228
49–1⅛	46–2	48–6	49–2	36–0	36–0
6.308 to 1	6.49 to 1	6.495 to 1	6.305 to 1.	6.33 to 1	6.33 to 1.
18–9	19–10½	19–10½	18–3	14–0	14–1½
10,467	9,700	10,875	10,500	5,705	5,765
770	741	816.64	775	435.3	439.8
.509	.51	.515	.510	.497	.502
19,006	17,806.6	20,585	19,110	11,428	11,510
50.41 19.52‖	37.43 18.2‖	48.89 20.096‖	46.41 19.0‖	32.31 16.14‖	29.6 15.8‖
598.72 19.52	666.01 18.2	695.3 20.096	613.06 19.0	539.6 13.14	599.8 15.8
198.63 19.52	217.97 18.2	261.1 20.096	198.62 19.0	176.75 16.14	167.4 15.8
135.3	91.44	Gauges out of order.	161.8	150.5	164.3
126.2 128.4	85.9 86.3	125.1 123.44	156.1 152.6	142.3 144.8	159.9 159.6
55.1 58.1	23.84 23.52	62.2 59.28	57.4 59.7	68.7 72.7	79.5 77.0
19.25 18.25	... ...	20.4 21.75	16.8 16.6	20 6 19.7	29. 28.5
125.79 123.83	115.35 113.95	116.42 116.08	127.4 126.7	157.98 153.94	51.5 153.3
754.78 743.65	692.10 683.7	814.94 812.56	849. 844.4	789.9 779.7	157.3 766.7
.55 .55	Wide. Wide.	Wide. 7.5 holes.	.8 .8	Wide. Wide.	Wide. Wide.
.69 .69 .69	.72 .67	69 .77 .77	.75 .76 .62	.76 .76 .76	.79 .78 .78
.69 .69 .70	.72 .67	.70 .77 .77	.72 .72 .65	.76 .76 .76	.79 .78 .78
51 37 25.3 9.93	42.20 14.55	50.17 21.87 9.52	58.8 25. 15.0	57.67 32.8 10.02	53.8 30.4 15.36
46.45 26.8 10.64	44.75 14.96	52.29 22.49 9 84	53.0 24. 13.6	55.65 35.5 10.72	53.1 28.7 15.0
30.32 30.68	26.54 26.83	28.23 29.20	38.23 35.28	33.61 34.94	37.26 36.1
25.7 26.1	26.17 26.06	24.5 24.3	25.9 25.6	24.84 25.02	24.1 24.
1.494	2.085	1.256	1.411	1 403	1.446
.365	.405	.677	.244	.229	.214
197. 131.	134.7 135.2	116.42 116.08	105. 125.	123. 120.5	102.8 101.8
66.48 41.23	21.98 20.87	... ...	29.9 31.6	60.88¶¶ 57.91¶¶	13.73¶¶ 13.59¶¶
199.7 201.25	343.3 313.9	177 177	208. 209.	123. 120.5	102 8 101.8
14.36 15.07	51.75 41.56	23.21 23.21	15.06 15.14	... ...	... ...
478.	486.	566.	665.	751. 763.	497.6
Eight: 110.67	Four: 75.57	Eight: 110.86	Eight: 117.32	Four: 52.03	Four: 34.7
3–4	3–4	4–6	4–6	4–6	4–6
2.03**	1.77	2.085	2.23	1.25	2.29**
1,590.06 1,427.92	1,426.61 1,395.45	1,682.65 1,749.17	1,353.5 1,223.0	519.76 495.41	464.7 464.0
1,603.63 1,700.43	... ...	1,514.10 1,555.69	1,355.2 1,316.6	588.78 631.89	524 7 502.0
1,594.39 1,664.29	1,732.92 1,761.08	1,625.62 1,680.61	1,759.6 1,574.3	471.16 498.28	688.7 670.10
4,788.08 4,792.64	3,159.53 3,156.53	4,845.59 4,985.48	4,468.3 4,113.9	1,579.7 1,625.58	1,678 1 1,636.10
9,717.86	6,452.22	9,877.49	8,674.07	3,324.37	3,341 6
42.85	45.84	28.81	53.	16.15	17.6
152.22	168.10	181.34	141.5	52.03	45.3
9,912.93	6,666.16	10,064.42	8,868.5	3,392.55	3,404.5
7.59 7.47	8.64 8.54	8.45 8.19	6.01 6.48	6.79 6.52	6.13 6.29
114.1	92.3	93. 93.	66.3	58.7 72.5	91.4
100.7 94.8	107.5 90.1	70. 73.	86.1 70.3	71. 84.1	89.3
63.7	65.3	56.	45.	40.4	36.
101.2 99.3	107.4 106.4	100. 97.	103.3 75.3	97.2 88.0	105.
109.5 103.1	116.4 115.1	129. 131.	112.4 78.3	91. 102.6	139.9
66,988.5 68,115.4	51,651.2 52,236.2	63,890. 65,920.	61,026. 56,489.	26,398 27,521.	27,692. 26,670.
59.02 60.01	53.9 54.5	61.2 63.15	47.98 44.41	72.88 75.98	52.06 50.1
1,163.19 1,182.76	943.4 954 08	1,117.6 1,153.0	1,156.3 1,070.4	1,037.5 1,083.0	1,044 1,005 6
9.14	10 16	8.69	8.93	7.70	8.03
21,774	21,875.	28,012.	26,071.	22,340.	23,000.
15.45	8.	18.62 18.38	20 3 19.87	17.19 16.07	19.9 20.8
...	20,096.32‖	30,353.32‖	21,590.5	...	9,402.1
...	46.07	44.9	40.0	...	42.7
...	1.290	1.767	1.290	...	1.145
...	3.01	3.15	2.43	...	2.76
2.03	2.33	1.707	1.887	2 385	2.4117
17.46	15.28	14.89	16.42	15.42	15.47
22.07	19.76	21.61	25.48	24 10	24.9
21.32	16.88	20 51	21.36	16.72	15.47
11.25	9.72	10.53	12.08	10.46	9.92
10.84	9.15	10.12	11.62	10.23	9.59
1 to 1.386	1 to 1.129	1 to 1.360	1 to 1.337	1 to 1.7	1 to 1.931
33.77 16.2	33.71 17.6	33.98 17.6	32.6 15.45	59.8 23.0	63.02 24.8

cluding water. ¶¶ Air, circulating, and bilge pump. ‖ Auxiliary boiler not used. * Aggregate I.H.P. ‡‡ Estimated. ** Closed ash-pits.

Dolphin.	*Petrel.*	*Vesuvius.*	*Chicago.*	*Boston.*
Six hours.	Four hours.	Eighteen minutes.	Six hours.	Six hours.
1,413	855	771	4,546	3,000
240	175	246–3	315	270
38	31–0	26–5	48–2½	42–0
7.5	5.66 to 1	9.32 to 1	6.53 to 1	6.43 to 1
23–9½	11–2	9–6	19–1	16–9½
4,717	3,852	4,574 ½	11,072	8,346
338	286	192	790	628
.5015	.494	.403	.547	.546
10,250	6,768	7,545	21,862	16,220
21.98 15.5‡	15.4 11.55‡‡	52.98 21.18‡	23.25 15.13‡	42.9 13.8‡‡
558.63 15.5	460.18 11.55	457.8 21.18	544.0 15.13	436.6 13.8
208.03 15.5	153.28 11.55	200.6 21.18	188.6 15.13	144.9 13.8
87.08	89.9	153.	87.3	87.6
85.21	79.	149.8 149.1	85.7 84.0	85. ½
28.04	28.	72.5 77.5	39.6 39-7	37. ½
...	... ...	28.17 ...	... ...	...
74.15	126.7	267.7 270.0	69.3 69.3	73.3
503.2	606.8	892.3 900.0	658.3 658.3	506.1
Wide.	Wide.	Wide. Wide.	Wide. Wide.	Wide.
.536 .602	.65 .65	.75 .74 .66	.66 .66	.66 .35
..	... ...	.75 .74 .66	.66 .66	... ...
41.41 12.95	51.3 13.4	58.2 33. 15.0	33.95 15.20	40. 14.9 16.37
...	... ...	63.1 28. 15.5	32.79 12.38	... ...
24.96	28.3	40.3 39.7	26.51 23.39	26.2
25.5	23.0	24.6 24.3	24.7 24.6	24.3
1.775	2.360	1.13	1.804	2.330
.199	.194	.148	.295	.128
30.71	126.7	267.7 270.0	44.5 43.6	26. ½
41.2¶¶	... ...	... ...	38.0 ½ 37.2 ½	81. ½
30 71	282.	264.5	44.5 43.6	26. ½
††	4.27	19.5 ½	... ...	...
...	660.	830.	500.	550
Four: 60. ½	Two: 35.	Four: 54.2 ½	Eight: 100	Six: 60. ½
6-0	4–6	4–6	4–2	4–2
Fire rooms open.	4.4	2.0	2.1	1.7 ½
1,021.95	504.2	570.0 624.8	1,068.9 1,031.32	1,395.6
...	... ...	673.6 576.5	... ...	...
1,122.22	496.	732.7 718.7	1,445.5 1,178.24	982.6 1,076.3
2,144.17	1,000.2	1,976.3 1,920.0	2,514.4 2,209.56	3,454.5
2,185.37	1,004.27	3,915.8	4,799.1	3,535.5
7.75	3.57	24. ½	††	††
60.0	36.94	58.0	284.6	244.1
2,253.12	1,044.78	3,997.8	5,083.7	3,779.6
8.73	7.94	5.69 5.90	8.7 9.88	8.0
87.3 97.1	96.6	97.	105. 120.	130. 125.
85.86	93.	90.	97.3 110.	125.
67	77.	63.	45.	73.
96.3	116.	90.	104. 107.3	116.
111.19	119.	122.	105. 102.	123.5
39,756.	21,142.	26,021. 25,634.	48,940. 43,092.	64,773.
43.72	55.	50.2 49.5	49.5 43.6	24.8
551.1	898.1	1,626. 1,602.	627.4 552.4	677.5
13.09	8.41	8.42	12.9	13.27
22,537.	18,750.	22,142.	21,452.	20,472.
11.65	24 5	14.8	9.9 9.9	20.4
4,926.	4,275	...	13,395.	11,770.
18.25	42.5	...	19.9	29.4
.752	1.528	...	.673	1.16
2.19	4.15	...	2.63	3.11
2.91	2.676	2,246.	3.91	2.72
8 34	12.56	20.5	7.56	9.45
11.758	17.15	51.12	11.5	...
11.71	15.8	23.58	9.6	11.5
5.867	8.21	16.8	4.86	5.32
...	7.54	16.05	...	5.29
1 to 1.64	1 to 1.134	1 to 1.988	1 to 2.167	1 to 1.15
33.5 27.13	24 5 11.7	108.2 24.1	17.08 14.8	25.75 21.46

pits. ¶ Main engines. ‡ Measured. ‡‡ Patent log. ½ Approximate. †† See auxiliaries.

XXXVII.

AUXILIARY ENGINES IN CONNECTION WITH THE MODERN MARINE ENGINE.

Abstract of a paper read before the Institution of Civil Engineers by Wm. Henry Allen :

Speaking of *circulating pumps* as being the first to be worked independently of the main engines the author says that the quantity of water used for circulating purposes has of course decreased with the reduction of water evaporated per I.H.P. per hour. Thus with from 25 to 30 pounds of steam per I.H.P. per hour the circulating pumps were required to give as much as 15 to 20 pounds for surface condensing, while with triple-expansion engines, using from 15 to 18 pounds per I.H.P., the regular quantity seldom exceeds 10 pounds.

The author has found the friction of the condenser tubes, together with the velocity of entrance, and the friction in the pipes to be equivalent to a head of from 5 to 10 feet. With this head, and a velocity in the pipes of 9 to 10 feet per second the efficiency has been found to be 50 per cent. In the Navy a velocity as high as 15 feet per second has been used, but the effect of this must be to considerably diminish the duty of the pump. The author recommends that for steamers having horse powers ranging from 6,000 upwards, the speed of the centrifugal pumps for circulating should not be less than 150 revolutions per minute, as a slower speed involves unnecessary bulk, the action of these pumps being perfectly certain at from 150 to 180 turns. In the Navy, during full power runs, their speed is increased to from 250 to 300 turns.

The custom of duplicating auxiliary engines seems now to have entirely fallen into disuse, past experience having proved it to be a totally unnecessary practice. If the circulating engines are well

made, there is no more occasion to duplicate them than the main engines.

As to the material employed in the construction of circulating pumps, all gun metal, all cast iron, or cast-iron casing and gun metal disc, the author has found that cast iron wore very well, but that cast iron and gun metal used in conjunction resulted in corrosion of the former.

Under the heads of *independent air pumps and auxiliary air and circulating pumps* nothing of importance is brought out.

Under the head of *fans for forced draft* the author states that owing to the great variety met with in the application of forced draft to boilers no general rule can be given governing the quantity of air to be supplied. An allowance of 260 cubic feet of air per pound of coal is a liberal allowance in most cases, but even this may, through leakage, fall short of the quantity desired in some cases.

The fans should be driven from the engines direct, and from 250 to 350 revolutions may be regarded as the safest and most satisfactory speeds for engines running on long voyages. The efficiency of the combined fan and engine is not high, the maximum obtained by the author on fans of from 4 feet 6 inches to 6 feet in diameter suitable for steamship purposes, being only 42 per cent. As, however, the power absorbed by the fan engines constitutes only a small fraction of the total power developed, it is well to provide large cylinder areas, so that the fan engines may on occasion be run faster than the average. For this reason it is well to calculate their size on the basis of an efficiency of 35 per cent.

From 40 to 50 feet per second is a safe velocity to allow in the suction orifice, as this permits considerable margin in the capacity of the fan. The ducts leading to closed ash pits should be of ample area and have none but easy bends; all angles and T pieces should be avoided. In several ships known to the author these passages were so constructed that no pressure was obtained in the ashpit, although the gauge at the fan registered 4 inches.

The author has experimented with double-inlet fans provided with a central diaphragm, but has found no benefit to accrue.

The trials proved that bosses having entrances curved to about the diameter of the inlet were in every way as efficient as if the diaphragm had been carried right through.

Of reversing engines the oldest type and one still in use for engines up to 1500 I.H.P is the "roundabout" engine. It consists of an ordinary reciprocating engine having a cylinder of from 5 to 8 inches diameter working, generally through means of a worm, a wheel carrying the crank pin, from which a connecting rod moves the reversing shaft or the links direct. The engine is reversed by a reversing valve. An advantage of this system is that the engine may be used for warming the main engines, by setting it going slowly, when it will alternately admit a small quantity of steam at top and bottom of the piston, as the links are shifted. The type of reversing engine now generally in use is of the direct type with hydraulic controlling cylinder and differential gear for holding the valve gear in the position indicated by the reversing lever by shutting off steam and closing the oil valves. To indicate any leakage an accumulator is used. It consists of a small cylinder with leather packed piston pressed by a steel spring, the piston rod passing through the top of the cylinder, and thus forming an index to its position inside. When any leakage occurs in the oil cylinder, the piston of the accumulator follows the leakage and keeps up the same pressure throughout the system. A few strokes of the hand pump, always supplied for reversing by hand, make good the loss by leakage. It is difficult to indicate a rule for the size of reversing cylinders, owing to the power required varying with the system of valves adopted. The author has fitted compound engines of 6,000 I.H.P. working at 100 lbs., piston valve on the H.P. cylinder, slide valve on the low, with a reversing engine of 20 inches diameter of cylinder by 24 inches stroke. For triple-expansion engines of 7,000 I.H.P., piston valves on the high and intermediate, slide valve on the low pressure cylinder, the reversing engine was 20 by 33 inches. For engines of 5,000 I.H.P., cylinders 14 by 20 have sufficed.

This same form of engine with hydraulic control is in use also for moving engine throttle valves.

For *dynamo engines* the double-acting engine, specially designed for driving dynamos, is the one in most common use now, after years of experimenting. Starting with very high speed of rotation, the number of revolutions of dynamos has been gradually reduced, until now 320 turns is the Admiralty standard, the dynamos being driven direct by an open twin double-acting engine, cranks opposite. The requirement is 33.7 lbs. of water per H.P. hour of output at the terminals of the machine. Until 1888 this quantity was from 40 to 42 lbs.; the gain in efficiency is due to increase in the efficiency of the dynamo and the use of higher pressure and compounding.

On merchant steamers, with the exception of some of the largest liners, nearly all vessels are now being fitted with single-cylinder direct-acting engines. The author thinks that these engines have been run at speeds somewhat too high, and recommends 270 revolutions per minute for engines capable of lighting boats requiring from 200 to 400 lights.

The application of fly-wheel governors to dynamo engines made these machines much more reliable and more easily managed. Regulation by a governor driven by belting is unsatisfactory owing to the slipping of the belt amid the grease and water of an engine room, and until the introduction of the fly-wheel governor it was necessary to watch the engine and dynamo in order to regulate, if necessary, its speed by hand. A fly-wheel governor will, in the event of the full load being suddenly thrown off, prevent the speed of the engine from being increased more than 3 per cent. The Admiralty specifications require that the speed shall be capable of hand regulation while the engine is running, for which purpose the fly-wheel governor is inapplicable, and the ordinary ball-and-spring form is generally employed, provided with hand regulation. It is a further condition in the Admiralty specifications that the momentary increase of speed and voltage, when the whole load is instantaneously thrown off, shall not exceed twenty-five per cent. of the normal.

The ship-lighting *dynamo* has been greatly improved of late,

and the drum-wound armature machine is one which has been found to work satisfactorily. The armature consists of a cast-iron hub mounted on a steel spindle, having laminated plates for the core, the whole being drum-wound on the outside.

The author recommends 60 volts for merchant ships, as being suited for both glow lamps and projectors.

The author's experience is that dynamos can be readily made with the cooling surface and the rate of development of heat so proportioned that neither armature nor field magnet rise more than 50 degrees beyond the temperature of the surrounding air after a run of six or seven hours. As temperatures in the engine room vary from 90 to 120 degrees, there can be no difficulty in making engines and dynamos fulfill their duties in as efficient a way as any of the other machinery in the engine room. Since the electric-light engines have generally to work harder and longer than any other in the ship, being in use both at sea and in port, their economy is a matter of some importance. The efficiency of ordinary ship-lighting dynamos, with double-acting engines, is considerable, in some cases as high as 83 per cent.; that of the ordinary mercantile form is over 81 per cent.

The discussion which followed turned almost entirely on dynamo engines and the application of electricity for driving all auxiliary engines on board, which latter plan found almost unanimous endorsement. Mr. Willans claimed for the multi-cylinder single-acting engine that it could be driven at high speeds with more safety and regularity than the double-acting engine, and he considered 320 revolutions as being rather above a safe speed for the latter for continued working at full pressure. He claimed for his engines that they were more easily managed and taken care of and that they were more economical, giving on an average an electric H.P. for 29.5 pounds of water per hour, using steam of 100 pounds pressure. With 130 to 140 pounds pressure the consumption of steam per electrical H.P. per hour was usually about 24 pounds; and with condensing engines using high pressure it could be reduced to 18 pounds per hour. Mr. Willans furthermore claimed great economy in oil used for lubricating over the double-acting engine, and less noise and vibration.

NOTES.

SUPERHEATING STEAM AND THE SERPOLLET BOILER.

Prof. A. B. W. Kennedy has recently made some experiments on a Serpollet boiler which go to show that great economy may result from the use of very highly superheated steam, attended with no difficulties in the operation of the engine due to the increased temperature of the steam.

The Serpollet boiler consists of separate coils of flattened steel tube, the width of the passage through the tubes being only $\frac{1}{32}$ inch. The coils are shaped very much like the tube of a Bourdon gauge with steam connection at one end, doubled back on itself. The feed water enters the lower coil and passes through the upper ones in succession. Upon starting, water is forced into the boiler by a hand pump, after which the engine takes up the work, just enough being injected at each stroke to supply the engine. The latter is stopped by diverting the water into a feed tank instead of into the boiler.

The mean boiler pressure was 54.7 lbs. per square inch, and the superheating in the steam chest was 194° Fahr. This would give a temperature of steam of about 500°, corresponding with that of saturated steam at a pressure of about 650 lbs.

The duration of the trial was 7 hours 54 minutes. The engine was of the vertical type and of ordinary construction, having a 5-inch cylinder with a 5-inch stroke. The revolutions were 284 per minute, and the indicated horse power 5.73. Steam was cut off at .6 stroke. The grate surface was 1.146 square feet and the coal burnt per hour per square foot of grate was 24.9 lbs. The theoretical evaporative power of the coal was 15.43 lbs. of water per lb. of coal. The feed water evaporated per minute was 2.43 lbs., which was equivalent to an evaporation of 6.97 lbs. of water per lb. of coal from and at 212° Fahr. The evaporative efficiency of the boiler was 45.2 per cent., 32.4 per cent. being carried off by the waste gases, and 22.4 per cent.

being lost by radiation. This is not a bad result for a boiler of such small size, and besides it must be remembered that the steam given off by this boiler is undoubtedly all steam and of unusual potency, containing as it does a large reserve of heat; so much so that the exhaust was superheated 55°. The coal consumed was 4.97 lbs. per I.H.P. and the feed water per I.H.P. per hour was 25.44 lbs., the latter result probably unprecedented for an engine of this size, and clearly indicating the value of superheating. There was no sort of inconvenience with the engine during the trial, and the amount of oil used for cylinder lubrication was not excessive. That ordinary engines can be run successfully for a length of time with steam as hot as is furnished by the Serpollet boiler is evidenced by a shop engine, which has been worked since April, 1890, at 9½ atmospheres with steam of 500°, including 140° of superheat. Prof. Kennedy found this engine in good condition, the cylinder walls being "smooth, glassy and quite greasy."

Some of the advantages touched upon in the report are: Perfect safety; ease of management, there being no anxiety as to water level; pressure kept constant with greater ease than with an ordinary boiler. Prof. Kennedy is of opinion that the Serpollet boiler would commend itself for use where only unskilled or intermittent attention could be given it. As to capacity, it was found that the evaporation was 10.1 pounds of water per hour per cubic foot of space occupied, the fuel being just under 25 pounds per square foot of grate per hour.

The difficulty one would expect in working a boiler with such restricted steam and water space as that of the boiler under discussion would be that arising through incrustation and choking. The boiler at Mr. Serpollet's works is cleaned out every week, and it has never become choked. As an experiment another boiler was purposely choked by using some very hard water. The cleaning process was then applied, which consists of pumping through the boiler a dilute solution of hydrochloric acid (one of acid to seven of water). It took about three-quarters of a minute working the pump before anything passed, and in about two minutes the coil was perfectly clear.—*London Engineering.*

THE HARVEY PROCESS OF SURFACE HARDENING STEEL.

Steel subjected to a decremental hardening under the patents of Mr. Hayward A. Harvey has been much heard of of late in connection with armor tests. In *Engineering News* of October 24 may be found a description of the process as far as made known by the patent specifications.

The armor plate of ordinary mild steel is laid, perfectly flatwise, upon a bed of dry sand or clay, deposited upon the bottom of a fire-brick cell or compartment erected within the heating chamber of a suitable furnace. The face to be treated is left exposed, and the compartment partly filled with a granular carbonaceous material, which having been rammed down upon the plate, is covered with a stratum of sand upon which is a layer of firebrick. The furnace is raised to an intense heat, about that required to melt cast iron, which is continued for such a period of time as may be required for the absorbtion by the metal near the upper surface of the plate of an additional one per cent. (more or less) of carbon, by virtue of which the surface metal acquires the capacity of hardening. The time required depends upon the efficiency of the furnace, the tightness of the packing, the intensity of the heat, and the depth to which the carburization is to extend. After carburization is complete the plate is taken from the furnace and, without removing the carbonaceous material from its surface, is allowed to cool down to the proper temperature for chilling. During cooling the carbonaceous covering protects the metal and prevents formation of scale, which, if present, would prevent proper hardening of the metal beneath it. Patches of the carbonaceous material may be removed at intervals and quickly replaced to discover whether the right temperature has been reached for chilling. This takes place when the metal is a dull cherry-red. Torrents of water are sprayed over the plate, or the latter is submerged and constantly moved.

The preceding process is that applied to armor plate. Guns are treated similarly, resulting in the formation of a hard inner skin adapted to resist erosion, and a deeper outer skin of hardened metal for increasing the strength.

The process described differs in no essential particular from

the cementation process for converting wrought iron into what is known as blister steel. The Harvey process, however, furnishes a product with a clean and smooth surface. This is believed to be due to the use of mild steel, a homogeneous metal, in place of wrought iron as raw material. Mr. Harvey, it is believed, considers the very intense heat employed an important factor. Closeness of contact between the metal and the carbonaceous material is another important point, as it is claimed that metal so packed will remain solid although exposed to a temperature exceeding its melting point.

THE EFFICIENCY OF STEAM SEPARATORS.

The following is copied from "Engineering News" of Sept. 12, 1891:

A test of the efficiency of six steam separators of different styles was recently made at Sibley College, Cornell University, under direction of R. C. Carpenter, Professor of Experimental Engineering. The use of separators, especially with high speed engines or with engines located at a considerable distance from the boiler, is becoming increasingly common; but very little is known as to the comparative efficiency of the different makes.

The separators tested were attached to a steam pipe supplying a 20 H.P. Straight Line engine; and to secure sufficient moisture in the steam for the purposes of the test, a section of the pipe was covered with a water jacket. The flow of steam was made uniform by keeping a constant load upon the engine. The test of each separator consisted in passing the steam for the engine through it, testing its percentage of moisture before entering the separator and after leaving it.

The percentage of moisture was determined in most cases by a universal calorimeter designed by Prof. Carpenter; but in a few cases a throttling calorimeter, designed by a student of Cornell University was used. The runs were each 20 minutes in length and the gauges and thermometers were read at intervals of 5 minutes.

The object of the tests was to determine the principles involved

in so constructing separators as to secure the greatest efficiency; and the names of the different separators tested were not published, the different styles being designated by the letters A to F. The tests showed that with such an amount of moisture in the steam as is common in ordinary practice, there was a great difference in the efficiency of the several makes. The following table shows the efficiencies of the various separators with steam containing various percentages of moisture:

Make of separator.	Quality of steam before.	Quality of steam after.	Efficiency per cent.
	DRIEST STEAM.		
B	97.5	99	60
D	96.1	97.4	33.3
E	98.1	98.5	21.1
F	97.7	97.9	8.7
C	95.6	95.8	4.5
A	98.0	98.0	0.
	STEAM WITH ABOUT TEN PER CENT. OF MOISTURE.		
B	87.0	98.8	90.8
A	90.1	98.0	80.0
D	89.6	95.8	59.6
C	90.6	93.7	33.0
F	88.9	92.1	28.8
E	88.4	90.2	15.5
	STEAM WITH ABOUT TWENTY PER CENT. OF MOISTURE.		
B	78.1	98.8	94.5
A	79.5	98.2	91.2
D	81.7	97.9	83.5
C	78.2	95.6	79.8
E	82.4	90.4	45.5
F	79.3	87.2	38.1

Of course a certain amount of the variations noted in the above tables were due to imperfections in the calorimetric measurements and variations in the quality of the steam. On the whole, however, the tests were conclusive in showing a marked difference in the efficiency of the different separators. The sum-

mary of the results was as follows, taking the different separators in the order of their efficiency:

Separator B.—With steam varying in quality from 66.1 per cent. to 97.5 per cent., the separator furnished steam varying from 97.8 per cent. to 99 per cent. The average efficiency was 87.6 per cent.

Separator A.—With steam varying in quality from 51.9 per cent. to 98 per cent., the separator furnished steam varying from 97.9 per cent. to 99.1 per cent. The average efficiency in 12 tests was 76.4 per cent.

Separator D.—With steam varying from 72.2 per cent., to .196 per cent., the separator furnished steam varying from 95.5 per cent. to 98.2 per cent. The average efficiency in 10 tests was 71.7 per cent.

Separator C.—With steam varying in quality from 67.1 per cent. to 96.8 per cent., the separator furnished steam varying from 93.7 per cent. to 98.4 per cent. The average efficiency in 12 tests was 63.4 per cent.

Separator E.—With steam varying in quality from 68.6 per cent. to 98.1 per cent., the separator furnished steam varying from 79.3 to 98.5 per cent., the separator seeming unable to remove the moisture very thoroughly from the samples of steam having more than 10 per cent. of moisture. The efficiency in 10 tests averaged 36.9 per cent.

Separator F.—With steam varying in quality from 70.4 per cent. to 97.7 per cent., the separator furnished steam varying from 84.1 per cent. to 97.9 per cent. The wet samples of steam were not thoroughly dried by this separator. The average efficiency in 10 tests was 28.4 per cent.

We quote as follows from the report of the tests as published in "Power":

"The volume of each of the separators was determined by filling with water, we expecting to find that to a certain extent, at least, the efficiency of separation depended upon the volume. We found no such relation existing, owing, undoubtedly, to the various modifying conditions arising from the several characteristics of the different forms.

"No marked decrease in pressure was shown by any of the separators, the most being 1.7 lbs. in E and ranging from this to 0.6 lbs. in B.

"Our investigation has led us to believe, and the results when taken in connection with the forms of separators clearly show that, although changed direction, reduced velocity and, perhaps, centrifugal force are necessary for good separation, still some means must be provided to lead the water out of the current of the steam.

"If such provision is not made momentary separation may occur; but before the water can drop or run from any surfaces in the direct current it will be again taken up by the rapidly moving steam which continually surrounds it. We believe that the high efficiency obtained from *B* and *A* was largely due to this feature. In *B* the interior surfaces are corrugated and thus catch the water thrown out of the steam and readily lead it to the bottom.

"In *A*, as soon as the water falls or is precipitated from the steam, it comes in contact with the perforated diaphragm through which it runs into the space below, where it is not subjected to the action of the steam.

"In *D*, the next in efficiency, this is accomplished by means of a >-shaped diaphragm which throws the water back into the corners out of the current of steam.

"In reality *E* depends for its action upon a pendent diaphragm placed across a greatly enlarged section of a horizontal pipe. With a good quality of steam this separator showed a very fair efficiency; it, however, seemed to be unable to handle steam with more than ten per cent. moisture. We believe that this inefficiency is due to the fact that the water has no means of escaping from the direct action of the steam.

"In *F* separation is claimed to be accomplished by centrifugal action. We have no doubt but that centrifugal action does take place, but there are no means provided for throwing the water beyond or out of the presence of the steam. Undoubtedly a constant whirl of mixed water and steam is maintained within the separator. Indeed, our engineer stated that he had steam

blowing freely from the drain while the water glass was full of water. This would seem to indicate pretty clearly the existence of a solid layer of water about the walls of the separator while the central portion was comparatively free. In our judgment this separator would become much more efficient were the inner surfaces provided with corrugations for breaking up this whirlpool-like action after the initial separation takes place.

"As a result of our investigation we have ample reason to conclude that steam separators may be constructed which will furnish a uniformly good quality of steam through as great a variation of condition as to moisture as will be found in any practice. We believe that we have fully established the principles to be followed in construction and operation of such a separator.

SHIPS.

UNITED STATES.

Torpedo Boat No. 2.—This boat was advertised a second time, no award of contract having followed the first bids. Two bids were received: one from the Cowles Engineering Co. of $117,490, the other from the Iowa Iron Works of $113,500. The latter firm has been awarded the contract, a board of officers having reported the establishment well equipped for the work. The Columbia Safety Steamship Co. sent a letter requesting that the time for opening the bids be extended. The vessel is to be completed in twelve months.

Cruiser No. 10.—The launch of this cruiser (for description see p. 253, Vol. 1 of the JOURNAL) took place Oct. 28. The ship was named the *Detroit.*

ENGLAND.

Second-class Cruisers.—A number of these vessels have passed their trials, and descriptions and data concerning them have been given in former numbers of the JOURNAL. At the risk of repetition, the following more detailed description is given from "Engineering" of Oct. 23, where it was published in connection with a report on the trials of the *Tribune* and *Thetis*, the last of three ships of this class built by James and George Thomson, the *Terpsichore* having been completed sooner. The Messrs. Thomson have succeeded in getting over 13 I.H.P. per ton weight of machinery in working order, with water in boilers and condensers.

The cylinders are 33½, 49, and 74 inches in diameter, with a stroke of 39 inches. The cylinders are separate castings and are connected together by steel stay rods. In addition the column heads have stout cast steel struts fitted between them. The cylinders are all steam jacketed, the working barrels of the H.P. and I.P. cylinders being of forged steel, those of the L.P. cylinders of hard, close-grained cast iron. The H.P. and I.P. cylinders have piston valves, the L.P. a balanced double ported slide; all have balance cylinders. The valves are worked by Stephenson links. The reversing engine is of the all-round type. The back columns are of cast steel, with guides bolted on, and the front columns are of forged steel. The main condensers are of naval brass riveted together; the total cooling surface is 10,000 square feet. There are two 14-inch Gwynne centrifugal pumps, the discharges from which are connected by an athwartship pipe, having sluice valves at each end, and also in the middle where it passes through the longitudinal bulkhead. The thrust blocks and collars are of cast steel; the latter are lined with white metal, and are of the horse-shoe pattern, each separately adjustable. Two auxiliary condensers, each with its independent air and circulating pump, have a combinded cooling surface of 1,000 square feet. In addition to the auxiliary machinery above specified there are in each engine room a set of air-compressing engines and reservoirs, electric light engines and dynamos, a main feed pump,

two fire and bilge pumps, a small pump for the drain tanks, and two evaporators and distillers with their pumps.

Steam at 155 pounds is furnished by five steel return-tube boilers, three double-ended, 13 feet in diameter by 18 feet 6 inches long; and two single-ended, 13 feet in diameter by 9 feet 7 inches long. The total grate surface is 573 square feet; the total heating surface 15,404 square feet. Each furnace has a separate combustion chamber, the draft for which may be independently controlled by dampers in the uptakes. For forced draft eight 5-foot fans are fitted capable of maintaining an air pressure of 3 inches to test the tightness of the whole space put under pressure, but the maximum allowed on the forced draft trial is 1¼ inches. The engine parts projecting above the protective deck are protected by five inches of armor on seven inches of teak backing round the engine hatchway between the protective and upper decks.

Each cruiser will carry 252 officers and men.

The following table gives the performances of the cruisers of this class that have had their trials, as far as they have become known:

Name.	Natural draft trial: duration, 8 hours; air pressure allowed, ½ inch.						Forced draft trial: duration, 4 hours; air pressure allowed, 1¼ inches.					
	Steam pressure at boilers.	Vacuum.	Revolutions.	I.H.P., total.	Air pressure.	Speed.	Steam pressure at boilers.	Vacuum.	Revolutions.	I.H.P., total.	Air pressure.	Speed.
Latona	153	27.6	136	7,247	.44	19.05	152	26.5	145	9,438	1.25	20
Melampus	147	27.5	135	7,656	...	...	146.7	26.5	140.5	9,093	1.28	18.35
Andromache	...	...	...	7,230	...	...	...	...	...	9,049	...	...
Apollo	...	...	129	7,488	.3	...	...	...	138	9,225	.82	...
Sirius	...	...	...	...	...	...	...	...	...	9,188	.80	19.35
Naiad	...	...	136.2	7,521	.4	...	...	...	145 5	9,250	...	19.25
Tribune	149	...	130.3	7,523	...	...	...	...	137.8	9,102	.60	...
Thetis	137.2	27.4	126.2	7,033	.55	...	149	27.0	136.2	9,496	.75	...

REMARKS.—The *Terpsichore*, also, has presumably successfully passed her trials, though the results have not become known. The first attempt resulted in failure. Under natural draft, after the vessel had been under way for about 7¼ hours, the tubes began to leak and a return to port was necessitated. The defects were repaired, and the following day the forced-draft trial was commenced. All went well for three hours, but then the tubes again commenced to leak, and the trial had to be discontinued. Examination developed the fact that not only had the tubes leaked, but a boiler seam had given out.

The *Sirius* is provided with single combustion chamber boilers; her success is doubtless due to the fact that the contractors were able to get the contract I.H.P. with only .8 inch of air pressure. The speed of 19.35 knots was obtained in a choppy sea. It is estimated that under favorable conditions a speed of 21 knots would have been reached.

The *first-class torpedo gunboats* provided for by the Naval Defence Act of 1889 will be similar to the *Sharpshooter* class (see p. 262, vol. 1 of the JOURNAL) with the displacement increased to 810 tons. The I.H.P. of the engines is fixed, however, at 3,500 in place of the 4,500 stipulated for in the case of the earlier vessels and which was never reached, 3,840 being the maximum ever attained, the corresponding speed being 19.75 knots in place of 21. The estimated speed for 3,500 I.H.P. is 19 knots. The boiler design has been modified and the engines allowed more weight, 17 I.H.P. per ton weight being exacted, whereas 24 I.H.P. per ton was realized on the *Karakatta's* trial.

One of these gunboats is to be built by Messrs. John I. Thornycroft & Co., of Chiswick. This vessel will be supplied with Thornycroft boilers in place of the marine locomotive type of boiler with which the others are fitted, and the I.H.P. is to be 4,500 instead of 3,500 as in the others.

Gossamer.—First-class gunboat, was selected for a series of progressive trials. Her draught was 8 feet 6 inches forward and 10 feet 6 inches aft. The following are the mean results: 111 revolutions gave 276 I.H.P. and 8.6 knots; 134½ revolutions gave 474 I.H.P. and 10½ knots; 165 revolutions, 856 I.H.P. and 12.6 knots; 191 revolutions, 1,314 I.H.P. and 14.7 knots; 214 revolutions, 2,027 I.H.P. and 16.3 knots, and 235 revolutions, 2,486 I.H.P. and 17.7 knots. On the full-power trial, using 1½ inches of air pressure, an I.H.P. of 3,600 and a speed of 19.8 knots were obtained, the boiler exhibiting signs of leakage at its conclusion.

Phoebe.—Third-class cruiser, has, after several unsuccessful attemps, successfully passed through her four hours' forced draft trial with the following results: Average speed, 19 knots; air pressure in fire rooms, 1.8 inches; mean boiler pressure, 149 pounds; pressure at engines, 146 pounds; vacuum, starboard, 25.2 inches; port, 24.5; revolutions, starboard, 155.9; port, 155.6; mean pressure in cylinders: high, 57.1 and 61; intermediate, 29.3 and 29.4; low, 14.9 and 14.9; I.H.P., starboard, 3,687, port, 3,848; total 7,535, or 35 in excess of contract requirement.

The *Phoebe* is a sister ship of the *Philomel* and the *Pallas* (see vol. 2 p. 583 and vol. 3, p. 268 of the JOURNAL).

Sandfly.—In consequence of the narrow escape of this vessel from foundering while cruising in the Bay of Biscay in a gale, the vessels of the *Rattlesnake* class are to be fitted with steel hatches and combings, and provided with relief valves for freeing the lower deck of water.

Blake.—It is stated that it has been decided not to submit this ship to a forced draft trial. Under these conditions the estimated 20,000 I.H.P. and speed of 22 knots will, of course, not be realized. It is said that the Admiralty will only demand 13,000 I.H.P., which will produce from 18 to 19 knots. The *Blake* has single combustion-chamber boilers, and although brick partitions have been erected, it is not considered safe to force the boilers to the intended maximum.

Vulcan.—After repeated failures this torpedo depot ship has just been put through an eight hours' trial of her machinery with fair success. Since her last steaming a filter had been fitted to prevent the oil and spume from passing from the auxiliary boiler into the condenser. The pressure in the stokehold was not allowed to rise above .7 inch, however, and the I.H.P. developed, 7,577, was little more than the estimated natural draft power, while the speed obtained, 17.2 knots, by patent log, fell short nearly a knot. It is improbable that any better results can be obtained from the present boilers; for although the trial with modified forced draft was pronounced a success, yet after its termination, and when the boilers began to cool down, the tubes of all boilers commenced to leak, while some of the tube-plates showed signs of buckling. After the ship has completed her under-water torpedo trials, the Admiralty will determine what is best to be done with her. Unless the naval authorities are satisfied to run her at three-quarters of her designed speed, it will be necessary to supply her with a new set of boilers.

FRANCE.

The Recent Naval Manœuvers.—The lessons to be learned from these manœuvers may be summarized as follows: Every squadron needs large cruisers for scouting duty. Only those of adequate displacement can make their full speed at all times irrespective

of the state of the sea. Small ships are not suited for this duty, as their speed must be much reduced in a heavy sea. The new cruiser *Tage*, a ship of 7,045 tons displacement, did excellent service, although crippled by an accident to her machinery, one of the cylinders, it is unknown which one, having developed a crack. Even so she made 16 knots with ease at all times.

For the reasons given "Le Yacht" thinks it advisable to give up exaggerated economy in ships' displacements and to fit them out with powerful and reliable engines. As regards the small cruisers the importance of giving them boilers of ample strength is emphasized.

The torpedo boats taking part in the manœuvers showed themselves to be unfit for use at sea. They were a constant source of care and worry for the Admirals, and impeded the progress of the slowest of the armor-clads. The limit of their endurance at sea is forty-eight hours, after which time they and their crew need the rest of a quiet anchorage.

"Le Yacht" proposes, therefore, that two torpedo transports, capable of carrying a total of 20 vedette boats be built. Only by this means will it be possible, the state of the sea permitting their use, to have at command reliable torpedo boats.

Davout.—A successful trial of this cruiser has not yet been accomplished. After the first attempt, last year, it was found necessary to make the ventilation more effective and to rebraze all the brazing of the steam pipes, which had been melted at the temperature of steam of 170 pounds. Later the hollow piston rods had to be replaced by solid ones and the valve travel shortened, because it was feared that the strains on the gear at full power would be excessive. These alterations completed, it was found on a trial made during July that the auxiliary machinery did not function properly, and the number of revolutions was less than that made on a natural-draft trial when a speed of 16.7 knots was obtained. There appears little doubt that if the power is developed the estimated speed will be reached, if not surpassed, but the engines and boilers are too light and too cramped.

Third-class Cruisers.—The six third-class cruisers, *Forbin*,

Surcouf, *Troude*, *Lalande*, *Cosmao* and *Coetlogon*, built practically on the same plans, are now completed, and all have had their trials, though those of the *Coetlogon* were to be repeated after certain alterations were made, one of which was the increase of the piston rods from 5.9 inches to 6.46. This trial has since been made, and the results are given below. All these ships have exceeded their contract speeds, but their engines are rather delicate and need constant attention. They should, however, be at all times good for from 17 to 18 knots without giving trouble.

The trial speeds reached were—

Forbin,	20.641 knots.
Surcouf,	20.510 knots.
Troude,	20.913 knots.
Lalande,	20.887 knots.
Cosmao,	20.603 knots.
Coetlogon,	19.700 knots.

On her second trial, made August 15, the last-named vessel attained a speed of 19.3 knots, .2 less than contract speed, on a two hours' run, and was accepted. The penalty was 100,000 francs for every tenth of a knot below contract speed.

Brennus.—This, the largest and most powerful battleship in the French navy, was recently launched. The following are some of the principal dimensions and data: length between perpendiculars, 361 feet; over all, 374 feet; breadth at water line, 67 feet; depth, forecastle to keel, 41 feet; maximum depth, 60 feet 7 inches; displacement, 12,000 tons; mean draught, 26 feet 3 inches. Two triple-expansion engines developing an aggregate I.H.P. of 13,500, drive twin screws, 17 feet 8 inches diameter. Estimated speed, 19 knots.

The hull, which has a straight stem without a ram, is constructed of mild steel and chromatized steel, except the stem and stern post, which are of wrought iron. The vital parts are protected from stem to stern by an armor belt projecting 35 inches beyond the upper cofferdam. It is formed of two strakes of steel, the first of which extends to a height of two feet above the water line. Its profile is a trapeze, the upper part of which measures 15¾ inches, the lower 13¾ inches. The second strake,

which has the same profile, tapers from 13¾ inches to 9.8 The belt rests on a teak cushion 19½ inches thick. The dead works are protected by 4.7-inch steel armor. The armored deck is 4.7 inches thick in two layers; all openings through it are protected by glacis. The battery is as far as possible protected against shell fire by numerous traverses. The lower cofferdam extends the whole length of the ship below the armored belt; and the upper cofferdam, which also extends completely round the ship, rises to a height of four feet above the main deck. The principal armament consists of three 13.38-inch 58-ton guns. Two of these guns will be placed in a pear-shaped central closed turret forward, and the third in a similar turret aft. The turret armor is 17.7 inches thick in front and 15.75 inches in rear. The turrets are mounted on hydraulic pivots and can be fired through a sector of 270° fore and aft. The secondary armament comprises ten 6.3-inch guns. Six of these are in a central double-staged redoubt, carrying 4 inches of armor, and the other four in circular closed turrets on the forecastle and quarterdeck. These turrets are on hydraulic pivots and carry the same armor as the redoubt. There are, besides, twenty quick-firing guns and revolving cannon, and five torpedo tubes, one at each angle of the lower stage of the redoubt, and the fifth right astern.

Hoche.—This battle-ship has recently been added to the French squadron in the Mediterranean (see p. 269, vol. 3 of the JOURNAL.) Although launched in 1886 the vessel was not fully commissioned until early in 1891.

The hull, from keel to water line, is of iron, but all that part above water is of mild steel. She is built on the cellular system, and her forward water-tight compartments are filled with cellulose. The principal dimensions are as follows: length on water line, 330 feet; beam, to outside of armor, 66 feet; draught, 27.25 feet; displacement (English tons), 10,412. There are two propellers actuated by four vertical compound engines. Under forced draft these developed 12,000 I.H.P. on trial, giving a speed of 15.7 knots. With natural draft an I.H.P. of 7,000 was developed. At a speed of ten knots the 600 tons of coal carried in her

bunkers should enable the ship to steam 4,000 knots. The boilers are designed for a pressure of 85.5 pounds per square inch above the atmosphere. Engines and boilers were built by contract at the engine works of Indret.

The *Hoche* is protected by an armor belt varying in thickness from 9 to 17.75 inches, extending the whole length of the ship, and an armored deck 3.01 inches thick.

The armament consists of two 13.39-inch 52-ton guns in turrets, one forward and one aft; two 10.63-inch 28-ton guns in barbette towers amidships on sponsons, one on each side. The turrets and towers carry 16 inches of armor. There are besides, fourteen 5.51-inch guns in battery, and four of the same calibre on the spar deck, two firing ahead and two astern.

Jaurèguiberry.—A contract for the construction of this battleship has been signed with the Société des Forges et Chantiers de la Méditerranée. The *Jauréguiberry* is of similar design to the *Lazare Carnot.* Like her prototype she will be fitted with three propellers. For description see p. 279, Vol. 3 of the JOURNAL.

Isly.—First-class protected cruiser was launched recently. Her principal dimensions are: length between perpendiculars, 346 feet; breadth, extreme, 43 feet 6 inches; draught, 19 feet 6 inches; displacement, 4,122 tons; I.H.P., 8,000; estimated speed, 19 knots.

Leger.—Torpedo gun boat was launched during August. She is a steel twin-screw vessel, 197 feet between perpendiculars, of 23 feet extreme breadth, and 10 feet 6 inches normal draught. The displacement is 450 tons. Engines of 2,200 I.H.P. are estimated to give her a speed of 18.5 knots.

Thornycroft & Co. have entered into an agreement with the Société des Forges et Chantiers de la Méditerranée for the construction in France of two torpedo boats similar to the *Conleur* recently built at Chiswick for the French government, and nearly identical with the boat recently built by the English firm for the Brazilian government. In the yard at La Seyne three somewhat larger boats than the *Conleur* class are also being constructed, likewise fitted with Thornycroft boilers. These boats are 143 feet long, 14.7 feet broad, and of 8.23 feet draught aft, with a speed of 20.5 knots at 104 tons displacement. There are two

boilers in each boat. Grate surface (one boiler), 29.8 square feet; heating surface, 1,825 square feet; weight complete with water, 9.83 tons; steam pressure, 200 pounds. The engine is triple-expansion, of about 1,200 I.H.P.; cylinder diameters, 17.3, 25.2, and 39.3 inches; stroke, 16½ inches. The first of these boats is at present undergoing her trials at Toulon.

The Société has further contracted to build for the French government a large, fast, seagoing torpedo boat called the *Mousquetaire*, which has a contract speed of 24½ knots, and will be fitted with two Thornycroft boilers. These boilers are among the largest constructed of the Thornycroft type, having a grate surface of 41.2 square feet, and weighing complete with water about 16 tons each. There are two sets of triple-expansion engines; diameter of cylinders, 15.8, 23.6 and 35.5 inches; stroke, 15.8 inches; collective I.H.P., 2,100. Ordinarily one boiler supplies steam to one engine, but the steam pipes are so arranged that either boiler can supply either engine or both engines. The length of the boat between perpendiculars is 154 feet; breadth, 15.4 feet; draught aft, 6.2 feet; displacement when fully loaded, 125 tons. She carries two 17.8-inch torpedo tubes on deck and two 37-mm. quick-firing guns, and is divided altogether into nine water-tight compartments.

ITALY.

Sicilia.—Battleship was launched recently. Length between perpendiculars, 400.3 feet; beam, 76.8; draught, mean, 28.7; displacement, 13,090 tons. There are two sets of engines with a collective I.H.P. of 19,500. Steam is furnished by 20 boilers. Estimated speed is 18 knots. The coal capacity is 1,200 tons.

Urania.—On a preliminary trial of this torpedo cruiser a mean speed of 19.6 knots was obtained. Without urging the fires excessively 20 knots was reached. The *Urania* is a modified *Tripoli*. Length between perpendiculars, 229.6; beam, 26.9; depth of hold, 17.83; draught, forward, 10.17; aft, 11.15; displacement, 831 tons. There is a protective deck, 1.57 inches thick. There are two three-bladed propellers. Estimated speed at 360 turns, 22 knots.

GERMANY.

Frithjof.—Second-class battleship was launched July 21. She is a sister ship of the *Siegfried.* Length, 239.4 feet; beam, extreme, 49.2; draught, mean, 17.7; displacement, 3,500; I.H.P., 4,800; speed, 16 knots.

There is a continuous armor belt. There are three 9.4-inch guns, two in a barbette turret forward, projecting beyond the ship's side, the third in a barbette turret aft. The turret armor is 7.5 inches thick.

RUSSIA.

Navarino.—Belted barbette battleship, under construction at the works of the Franco-Russian Company, St. Petersburg, is of the following dimensions: Length, 338 feet; beam, 67 feet; displacement, 9,476 tons; I.H.P., 9,000, which is calculated to give a maximum speed of 16 knots per hour. She will be provided with twin screws. The total coal capacity is given as but 700 tons.

The armor on her partial belt is 16 inches thick, and that on each of her barbettes is 12 inches. Each barbette will mount two 12-inch 52-ton guns; in the central box-battery there will be eight 6-inch guns, and on the hurricane deck there will be ten quick-firing guns. There will be six topedo tubes.

Gremyastchy and *Otvajny.*—Belted gun vessels. Length 225 feet; breadth 41 feet; displacement 1,500 tons; twin-screw engines of 2,000 I.H.P. are to give a speed of 15 knots. There will be a complete steel belt varying from 3 to 5 inches in thickness. Armament, one 9-inch gun, one 6-inch gun, eight quick-firing guns, and two torpedo tubes.

Rurik.—The belted cruiser *Rurik* is under construction at the Baltic Works. She is 426 feet in length and 67 feet in breadth, and her displacement is about 11,000 tons. She will be the largest cruiser afloat. Four engines, developing 15,000 I.H.P. driving twin-screws, are estimated to give a sea speed of 18 knots. Along four-fifths of the water line there will be a 10-inch belt. The protective deck is 2½ inches thick. She will carry enough coal to steam 20,000 nautical miles at her economical speed of

boilers in each boat. Grate surface (one boiler), 29.8 square feet; heating surface, 1,825 square feet; weight complete with water, 9.83 tons; steam pressure, 200 pounds. The engine is triple-expansion, of about 1,200 I.H.P.; cylinder diameters, 17.3, 25.2, and 39.3 inches; stroke, 16½ inches. The first of these boats is at present undergoing her trials at Toulon.

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10 knots. The armament will consist of four 8-inch guns, sixteen 6-inch, fourteen 4.7-inch, eighteen small quick-firing guns, and five torpedo tubes.

Gangut.—The belted cruiser *Gangut* will have a displacement of 6,590 tons. Her length is 287 feet; beam, 62 feet; I.H.P., 9,300, which should give a maximum speed of 16½ knots. She has a partial belt 16 inches thick. The armament is not decided on.

SPAIN.

Emperador Carlos V.—One of six first-class armored cruisers building. Displacement, 9,235 tons; greatest mean draught, 25.75 feet; I.H.P., 15,000; speed under natural draft, 19 knots; under forced draft, 20 knots; radius of action, 12,000 nautical miles.

Armament: Two 11-inch guns in turrets; eighteen rapid-fire guns of different calibers; six revolving cannon and six torpedo tubes.

The ship will have a protective deck, and sides protected by armor 2 inches thick, in two layers, one of Siemens-Martin, the other of chrome steel. The turret armor will be 9.8 inches thick, either all steel or compound. Forward and aft there will be thwartship bulkheads 52.5 feet long, 5.9 feet high, and 2 inches thick.

The time allowed for building is 40 months.

Viscaya.—Cruiser of 7,000 tons displacement, was launched recently. She is the second of three similar vessels building. Length between perpendiculars, 340 feet; breadth, 65 feet; normal draught, 21 feet 6 inches. The protective deck is 2 inches thick in the middle and 3 inches on the slopes. The two barbettes are cased with armor 10.5 inches thick. Two sets of triple-expansion engines are designed to develop 13,000 I.H.P.; estimated speed, 20 knots.

Nueva España.—Torpedo gunboat. This vessel is about to begin her trials. Principal dimensions: Length between perpendiculars, 190 feet; beam, extreme, 23 feet; displacement, 571 tons; draught, mean, 9 feet.

Twin screws are driven by two sets of vertical triple-expansion engines of an aggregate I.H.P, under forced draft, of 2,600

at 210 revolutions. With natural draft 1,600 I.H.P. is to be developed. The coal consumption under forced draft is limited to 2.25 lbs.; under natural draft to 1.78 lbs. per I.H.P. per hour.

The cylinder diameters are 21, 30 and 46 inches; stroke, 21 inches. There are eight ejectors in the vessel, located in the engine and boiler compartments and in the compartments at the bow and stern, and capable of discharging 130 tons of water per hour. The centrifugal circulating pumps are capable of discharging 160 tons per hour. The fire and bilge pumps, the four auxiliary feed pumps, and two hand pumps, add 64 tons more.

There are four boilers in four separate compartments. The two after ones are two-furnace Scotch boilers; the two forward ones are of the locomotive type. Total heating surface, 4,100 square feet; steam pressure, 130 pounds. The tubes are of steel, 2½ inches in diameter for the Scotch boilers, and 1¾ for the locomotive boilers. The stop valves may be worked from deck, as also valves for injecting steam into the furnaces to deaden the fires in case of accident. Air pressure to be carried, 2½ inches. Ashes may be hoisted while the fire rooms are under pressure.

The weight of all machinery is 160 tons; coal capacity, 125 tons.

CHILI.

Presidente Pinto.—Protected cruiser, sister ship of the *Presidente Errazuriz*. The following are some of her principal dimensions: Length, 268 feet 4 inches; beam, 35 feet 9 inches; draught, mean, 14 feet 7 inches; draught, aft, 16 feet 5 inches; depth, 23 feet 4 inches; displacement, 2,081 tons. There is a protective deck varying in thickness at the sides from 1.57 to to 2.36 inches, in the middle from 1.18 to 1.38. The ships are sheathed in teak, Canadian elm, and copper, the sheathing being carried well up above the water line. There are two masts with military tops, and a conning tower with 3.15 inches of armor.

The propelling machinery consists of two sets of horizontal triple-expansion engines, in separate compartments. They have developed on trial 3,560 I.H.P. with natural draft and 5,500

with forced draft, the speeds corresponding being 17 and 18½ knots. The air, feed, and circulating pumps are driven by a two-cylinder vertical compound engine for each engine room. Four locomotive boilers, each with three corrrugated furnaces, supply steam at 160 lbs. pressure. The normal coal supply is 170 tons, but 230 tons can be carried in time of necessity. On 170 tons the ship can steam 1,446 nautical miles at 15 knots, and 2,550 at 12 knots; on 230 tons these distances become 2,550 and 4,500.

The armament consists of four 5.9-inch Canet quick-firing guns in sponsons; two 4.7-inch guns, one on the forecastle, the other on the poop, and a number of smaller quick-firing guns and revolving cannon.

BRAZIL.

Torpedo Boats.—The second of two torpedo boats built by J. I. Thornycroft & Co. for the Brazilian government has lately had her trials. These boats are very similar to those built recently for the Argentine government. Length, 150 feet; beam, 14 feet; draught (greatest), 5 feet 3½ inches.

The boats are fitted with Thornycroft's patent system of double rudders. These are placed side by side in an athwartship line, and arranged so as to partly surround the screws. The rudders do not trail in a fore-and-aft line, the after edges being closer together than the forward ones. The most desirable angle to be formed by the planes of the rudders was found by preliminary experiments upon the measured mile, a number of runs being made with different angles, until it was found that the maximum speed was attained. When properly set in this way, the rudders contribute a slight propelling effect, which is sufficient to counterbalance their own surface resistance. The high velocity of the water passing between them causes a reduction of pressure on the internal surfaces, and the excess of pressure on the outsides, acting upon the inclined surfaces, gives a small component of thrust in the direction of the advance of the vessel.

The machinery consists of two sets of triple-expansion engines driving twin screws. Steam is furnished by two Thornycroft

boilers; aggregate heating surface, 4,132 square feet; grate surface, 75.5 square feet.

The builders had guaranteed a speed of 25 knots on the measured mile and 24 knots on a two hours' continuous trial, the boiler pressure being limited to 200 lbs. The first boat, tried May 26th, made, under these conditions and with a load of 19 tons on board, 25.086 knots on the mile and 24.4 knots on the two hours' run. Permission was then granted to try the second with a boiler pressure of 210 lbs., and the speeds attained were 25.858 and 25.387 on the mile and for two hours respectively. The air pressure was from 1½ to 2 inches; the I.H.P. developed, 1,800. The fuel economy was measured on a run of six hours at 10 knots, and 1.393 lbs. of coal per I.H.P. per hour were burned. This trial was run with one engine, the propeller of the other being uncoupled.

MERCHANT STEAMERS.

The Atlantic Record.—During the past season the Atlantic record, previously held by the *City of Paris*, has been repeatedly lowered, first by the White Star Liner *Majestic*, and then again by her sister ship, the *Teutonic*. The following table gives the performances of these three ships on their record-breaking trips:

	Teutonic.	*Majestic.*	*City of Paris.*
First day,	460	470	462
Second day,	496	501	493
Third day,	505	497	502
Fourth day,	510	501	506
Fifth day,	517	491	509
Sixth day,	290	317	316
	2,778	2,777	2,788
Time,	5d. 16h. 3m.	5d. 18h. 8m.	5d. 19h. 18m.
Averge speed,	20.36	20.1	20.01.

The *Teutonic* also holds the record for the best day's run with

517 miles against 515 for the *City of Paris;* this is an average of 20.8 for the twenty-four hours. The speed of the *Majestic* on her record-breaking trip was less than that attained on a previous voyage, when she averaged 20.23 knots, but ran over a longer course. The *Teutonic* now also holds the record for the fastest eastward trip, her time being 5d. 21h. 3m. against 5d. 22h. 55m. of the *City of Paris* and *City of New York.*

Fuerst Bismarck.—A brief description of this steamer may be found on p. 289, vol. 3 of the JOURNAL. As several inaccuracies occur in that description it will be well to repeat a portion of it here. The principal dimensions are: length on water line, 502 feet 6 inches; breadth, 57 feet 6 inches; depth of hold, 38 feet; displacement at 26 feet draught, 12,900 tons.

All the cylinders are steam jacketed. The H.P. cylinders have piston valves, the I.P. and L.P. slide valves; the L.P. cylinders have two slides each, working side by side. The cylinder diameters are: H.P., $43\frac{5}{16}$ inches; I.P., $66\frac{5}{16}$ inches; L.P., $106\frac{5}{16}$ inches; stroke, 63 inches. The contract I.H.P. was 14,000 and the speed 19. On the trial, during which the engines were run at full power for twelve hours, the maximum I.H.P. was 16,412, and the speed, measured for two hours, 20.7 knots.

The condensers are at the back of the engines, and carry facings for the columns, being themselves part of the bedplates. Each condenser is divided into two separate parts, the exhaust passing equally into both. There are three single-acting air pumps in each engine room, driven by a three-cylinder simple engine. This engine also drives a bilge pump and a pump for supplying cooling water to the different bearings. Each set of air pumps can draw from the condenser of either engine, and is large enough to do the work for both main engines. The centrifugal circulating pumps are similarly arranged, *i. e.*, they can discharge into either condenser and are capable of supplying both.

The crank shaft is made in three built-up, interchangeable sections, and, together with the other shafts, is made of crucible cast steel.

The main boilers are arranged in three groups, with athwartship fire rooms. One auxiliary boiler is located in the forward

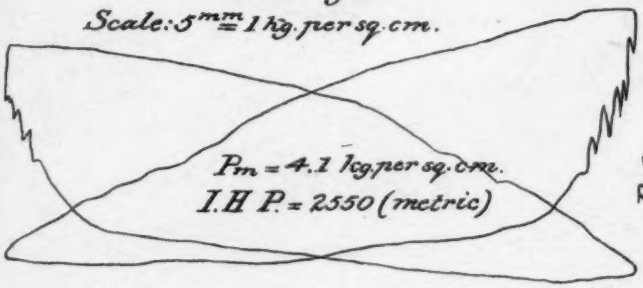
Indicator Cards of the 'Fuerst Bi

April 25th 1891.

Port Engine.

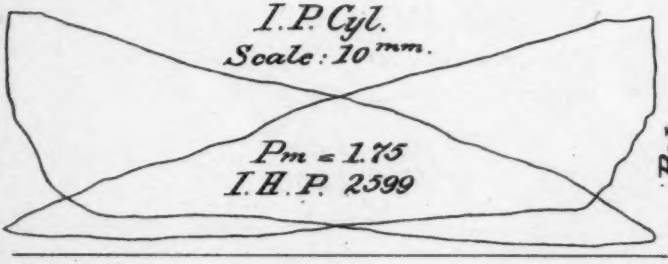
H. P. Cyl.

Scale: 5 mm = 1 kg. per sq. cm.



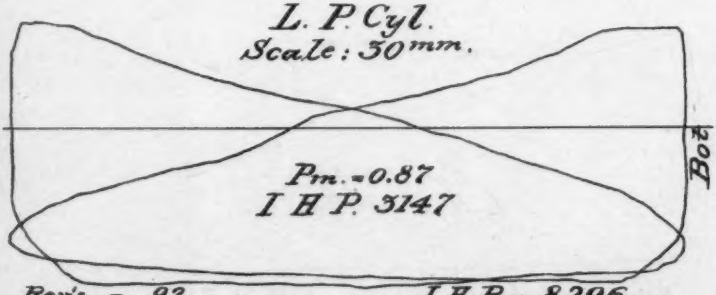
I.P. Cyl.

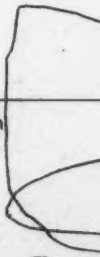
Scale: 10 mm.



L. P. Cyl.

Scale: 30 mm.





Rev's. = 92 *I.H.P. = 8296*

Revs.

Total I.H.P. 16,076

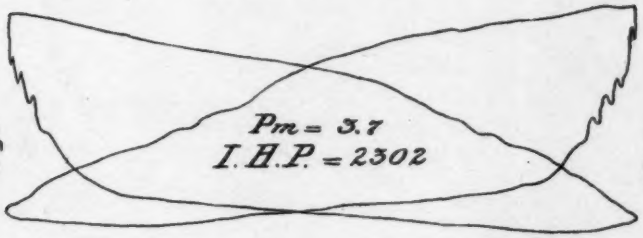
Cut-off { *H.P. = 0.68* / *I.P. = 0.60* / *L.P. = 0.60*

uerst Bismarck'.
891.

Starboard Engine.

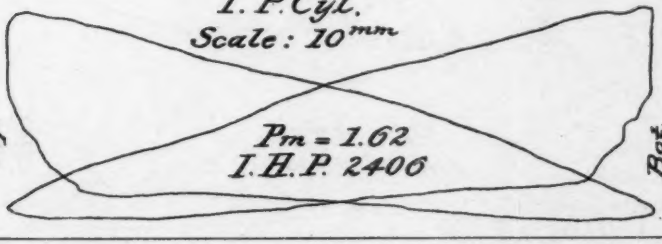
H.P.Cyl.

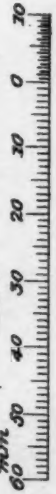
Scale: 5 mm



I.P.Cyl.

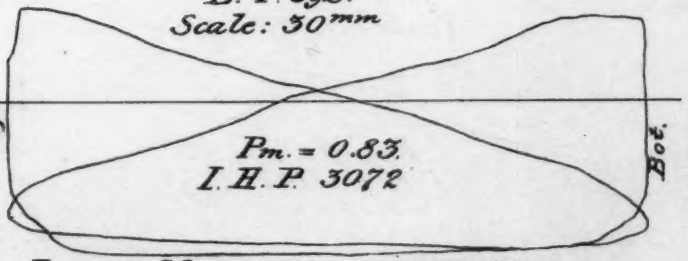
Scale: 10 mm





L.P.Cyl.

Scale: 30 mm



Revs. = 92. I.H.P. 7780

6076
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60

part of the middle fire room. The main boilers are 15 feet 7 inches in diameter by 19 feet 1 inch in length. Each main boiler has eight corrugated furnaces of 39⅜ inches mean diameter. The total grate surface of the main boilers is 1,450 square feet; the heating surface, 47,000 square feet. The tubes are 3 inches external diameter and 7 feet 5 inches long. The working pressure is 157 pounds. There are fans for ventilating the stokeholds, but the boilers are designed for natural draft. The bunker capacity is 2,700 tons.

The steam pipes are of copper and are wrapped with steel wire.

On her first trip to New York the *Fuerst Bismarck* made the best maiden passage, her time being 6 days 14 hours 7 minutes from Southampton to New York, the average speed being 19.5 knots. Her best day's run was 498 miles, being at the rate of 20.1 knots. On the homeward trip she beat the eastward record, reaching Southampton in 6 days 13 hours 29 minutes, her average speed being 19.78 knots.

The indicator cards facing p. 610 were kindly furnished by the Chief Engineer of the ship, Mr. L. Jonas, and transmitted by Mr. Thos. H. Barrett, U. S. Local Inspector of Steam Vessels. On another set, not published, the coal consumption was given as 1.7 pounds per I.H.P. per hour, and as 255 tons for twenty-four hours. The per cent. of refuse was 12½. The slip of the screw was 16 per cent., the ship making 19 knots on 85 turns.

La Touraine.—Mention was made of this, the latest steamer added to the fleet of the Compagnie Générale Transatlantique, on p. 362, Vol. 1, of the JOURNAL, and a description given on p. 432, Vol. 2. Following is more detailed information: length over all, 538 feet 9 inches; between perpendiculars, 517 feet 7 inches; breadth, extreme, 56 feet 2¾ inches; draught, loaded, 23 feet 4½ inches; displacement at this draught, 11,685 tons. The double bottom tanks are 4 feet 3 inches deep and have a capacity for 979 tons of water ballast.

The engines occupy 43 feet 11 inches of the vessel's length. They are separated by a midship bulkhead extending to the water line, above which the division is carried up to the skylight by a passage completely closed in. The cylinders are all steam jacketed. The high and intermediate pressure cylinders have piston

valves—the low slide valves. All the levers, connecting rods, and shafting are of mild steel, while the frames, condensers, bedplates, pistons, and cylinder covers are of cast steel. There are two condensers to each engine. They are located between the outboard frames of the cylinders, and are seated on the bedplates. They are 9 feet 2¼ inches long, of the same height, and 5 feet 7 inches wide. The tubes are ⅝ inch inside. The total cooling surface is 22,445 square feet. Each condenser has its own independent circulating pump and its own air pump, driven from the main engine. The air pumps are 37¼ inches diameter and 25⅝ inches stroke. The propellers are 19 feet 8¼ inches in diameter, and have a pitch of 29 feet 6 inches. They have steel bosses and three gun-metal blades. Steam is supplied by nine boilers in two groups in separate compartments. The forward group contains three double-ended, the after group three double-ended and three single-ended boilers. The D. E. boilers are 20 feet 4⅛ inches long, the S. E. 10 feet 6⅛ inches; the diameter of all is 14 feet 9⅛ inches. The tubes are 7 feet 8½ inches long and 2⅛ inches internal diameter. The D. E. boilers have six furnaces each, the S. E. three; they are corrugated and of 3 feet 11¼ inches least internal diameter. The grates are 6 feet long. The total grate surface is 1,162½ square feet, the heating surface, 33,617 square feet. The coal consumption is given as 252 tons a day for the main engines and eight for the auxiliaries. Forced draft is employed, there being four fans in each stokehold. Those in the larger compartment are capable of furnishing 23,540 cubic feet a minute, those in the smaller compartment 14,714 cubic feet. The ship is lighted by electricity, there being 300 lamps of 16-candle power, and 572 of 10-candle power. Current is supplied by two dynamos driven direct at 380 turns by compound engines. The engine and dynamo shafts are separate, and connected by an elastic coupling. A third dynamo is held in reserve.

On her trial the *Touraine* made 18.75 knots under natural draft with 9,500 I.H.P. and 72 revolutions, and 20 knots under forced draft with 12,000 I.H.P. and 78 revolutions. The best average speed to New York has been 18⅛ knots, the highest for twenty-four hours being 19½.

OBITUARY.

CHIEF ENGINEER JAMES BUTTERWORTH, U. S. N., was born in England, March 31st, 1840; came to Massachusetts when about ten years of age; was educated in the public schools of Boston, taking a private course in mechanics and draughting; learned the trade of machinist; was appointed a Third Assistant Engineer May 23d, 1861, and served on board of the *Susquehanna* through the battles of the Gulf Squadron; was promoted to Second Assistant October 29th, 1862, and served on board the *Niagara* in her cruisings after the *Alabama;* was promoted to First Assistant Engineer July 25th, 1866, and after a short tour of duty at the Boston Yard he made a cruise in the *Dakotah;* he was transferred to the *Saginaw*, and was in that vessel when she was wrecked on Ocean Island, October 29th, 1870. It was here that Mr. Butterworth secured a quantity of iron pipe from the wreck, which he improvised into a distilling apparatus, but for which the crew would have been without water to drink. With the assistance of an officer and a few men of the crew he put a deck on the gig, and fitting her sails she carried Lieutenant Talbot and crew to the Sandwich Islands (about 1,500 miles to leeward), where news of the wreck was sent to the United States. From the effects of the exposure on Ocean Island Mr. Butterworth never fully recovered. His next duty was in the Boston Yard; then a cruise in the *Narragansett* in the South Pacific. He was promoted to Chief Engineer March 10th, 1881, since which time he has made a cruise in the *Marion*, Asiatic station. A modest, unpretentious man, painstaking and conscientious in the execution of his duties, Mr. Butterworth made himself a popular and respected shipmate. A gentle, exemplary man, he was a kind father and affectionate husband.

CHIEF ENGINEER GEORGE H. WHITE, U. S. N., who died February 23, 1891, was a native of Pennsylvania. He entered the Navy as Third Assistant Engineer in 1858 and resigned in 1859. He re-entered in 1861 and served as Third Assistant on the *Colorado*

and the *Monitor*. He was promoted to Second Assistant in 1862 and served on the *Catskill*, *Nantucket* and the *Ottawa* in succession. Promoted to First Assistant in 1865, after which he made a number of cruises, doing Chief Engineer's duty on some of the iron clads, and had shore duty at the Navy Yards at Mare Island and Philadelphia, and at New York. After his promotion to Chief Engineer, November 23, 1878, he made cruises in the *Wachusett*, the *Powhatan*, and *Galena*, and was on special duty at Philadelphia, Brooklyn, and Norfolk. He was retired November 18, 1890, for disabilities incidental to the service.

Mr. White was a thorough engineer and an expert on the manufacture of iron and on cupola practice.

HIRAM HAINES was born in Virginia. He entered the naval service as Third Assistant Engineer February 16, 1852, which rank he held at the time of his resignation in the latter part of 1854. At the time of his death Mr. Haines was engaged in mining engineering at Cordova, Walker County, Alabama.

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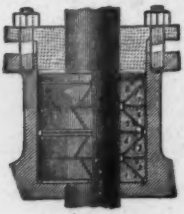

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